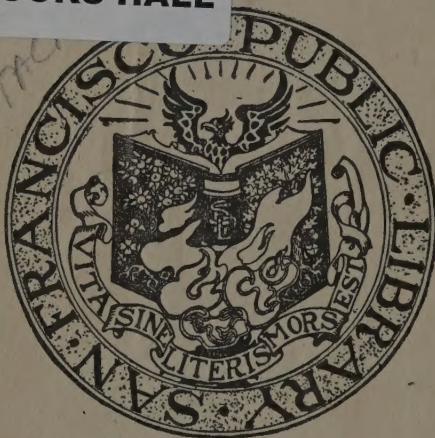


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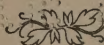
MODERN ELECTRIC RAILWAY PRACTICE

VOLUME III

POWER STATIONS
STEAM AND GAS ENGINES
GENERATORS AND
SWITCHBOARDS

BY
CALVIN F. SWINGLE
and a Corps of Practical Electrical Experts

ILLUSTRATED



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PREFACE.

Although the electric locomotive and motor car are rapidly taking the place of the steam locomotive for transportation purposes, and the time no doubt is not far distant when the power generating machine known as the steam locomotive, laboring under the disadvantage of having to carry its fuel and water with it, will be a thing of the past, the fact still remains that the power must be generated. There must be a prime mover, in and through which heat, that form of energy which may be applied to or taken away from bodies, may be utilized for the generation of power. The central station, with its modern appliances for the economical consumption of fuel and production of heat, and its machines of various types, such as steam engines, gas engines, etc., by means of which this heat is transformed into energy or power, capable of turning the dynamo, which, in turn, transmits this power where it is needed to do the work, will be the subjects discussed in the pages following.

The author proposes to deal mainly with the operation of the aforesaid machines, rather than the construction, and erection of the same, although their design and construction will receive some attention also.

CENTRAL POWER HOUSES AND SUB-STATIONS.

STEAM BOILERS.

The types of boilers most commonly used in central stations are, first, the horizontal return tubular boiler, and, second, the water tube boiler. The first mentioned



type consists of a shell, having tubes of small diameter (2 to 4 in.) extending from head to head, through which the hot gases, after having passed under the bottom of the shell, return on their way to the stack. Fig. 1 shows a general view of a horizontal tubular boiler and setting. Although this form of boiler is probably not so economi-

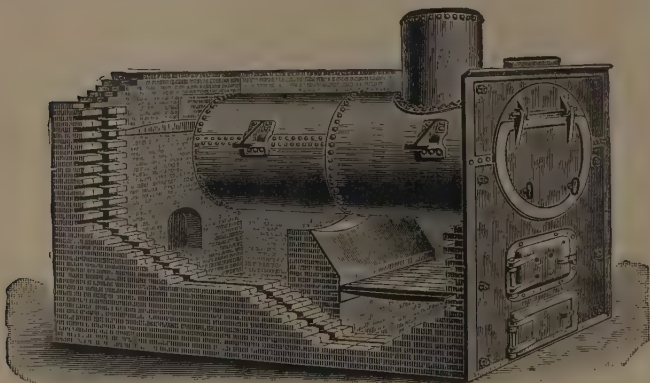


Fig. 1. Standard Horizontal Boiler with Full-arch Front Setting.

cal in the utilization of the heat contained in the fuel as are some other types, still, with the proper setting and care in operation, fairly good results may be obtained.

THE WATER TUBE BOILER.

In this type of boiler the water is carried in tubes 3 to 4 in. in diameter, sometimes vertical and sometimes inclined, and connected at the top to one end of a steam drum, and having the lower ends of the tubes connected

to a mud drum, which is also connected to the opposite end of the steam drum, thus providing for a free circulation of the water. Of the latter type there have been many different kinds evolved during the last one hundred years, the majority of them having had but a brief existence, being compelled to obey the inexorable law of the survival of the fittest, and today there are a few excellent types of water tube boilers which have become standard and are being extensively used. The margin of safety as regards disastrous explosion appears to be in favor of the water tube boiler. It is not contended that they are entirely exempt from the danger of explosion. On the contrary, the percentage of explosions of water tube boilers, in proportion to the number in use, is probably as large, if not larger, than it is with boilers of the shell or return tubular type, but the results are seldom so destructive of life or property, for the reason that if one or more of the tubes give way the pressure is released and the danger is past.

REQUIREMENTS OF A PERFECT STEAM BOILER.

The late Mr. George H. Babcock, of Plainfield, N. J., ex-president of the American Society of Mechanical Engineers, was probably a closer student of steam boiler design and construction than any other engineer who has lived during the present century. Shortly before his death he wrote that the result of his engineering experience and scientific investigation for a period of over fifty years in the field of steam boiler practice had established twelve requirements of a *perfect* steam boiler.

1st. The best materials sanctioned by use, simple in construction, perfect in workmanship, durable in use, and not liable to require early repairs.

2d. A mud drum to receive all impurities deposited from the water in a place removed from the action of the fire.

3d. A steam and water capacity sufficient to prevent any fluctuation in pressure or water level.

4th. A large water surface for the disengagement of the steam from the water in order to prevent foaming.

5th. A constant and thorough circulation of water throughout the boiler, so as to maintain all parts at one temperature.

6th. The water space divided into sections, so arranged that should any section give out, no general explosion can occur, and the destructive effects will be confined to the simple escape of the contents; with large and free passages between the different sections to equalize the water line and pressure in all.

7th. A great excess of strength over any legitimate strain; so constructed as not to be liable to be strained by unequal expansion, and, if possible, no joints exposed to the direct action of the fire.

8th. A combustion chamber, so arranged that the combustion of gases commenced in the furnace may be completed before the escape to the chimney.

9th. The heating surface as nearly as possible at right angles to the currents of heated gases, and so as to break up the currents and extract the entire available heat therefrom.

10th. All parts readily accessible for cleaning and repairs. This is a point of the greatest importance as regards safety and economy.

11th. Proportioned for the work to be done and capable of working to its full rated capacity with the highest economy.

12th. The very best gauges, safety valves and other fixtures.



Fig. 2. Front View of 250 H. P. Cahall Horizontal Boiler.

THE CAHALL WATER TUBE BOILER.

Figures 2 and 3 show, respectively, front and side views of the Cahall horizontal water tube boiler. These boilers are built in sizes of from 125 horsepower up to 850 horsepower, in single units. The boilers are built for working pressures of from 160 pounds per sq. in.

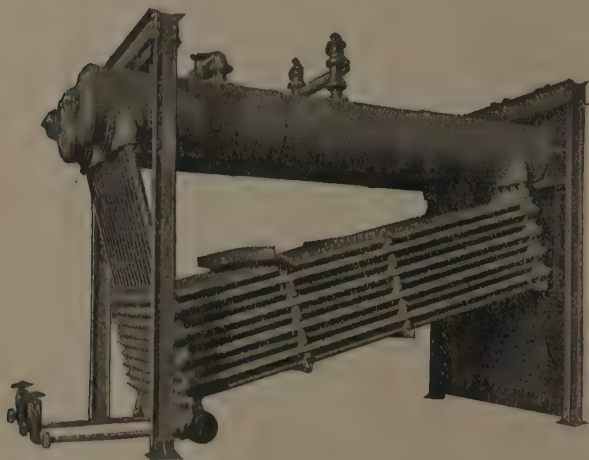


Fig. 3. Side View of 250 H. P. Cahall Horizontal Boiler Suspended Ready for Brick Work.

up to 500 pounds per sq. in. The steam drums are made of the best open hearth flange steel, the heads for the drums being of the same material, hydraulically flanged. All the sheets are beveled on the edges, bent into shape, and the rivet holes drilled after bending. This insures absolutely round holes, without crystallization, and allows calking of all seams, both inside and

outside. In boilers, the working pressure of which is not to exceed 160 lbs. to the sq. in., the longitudinal seams in the drums are double riveted. In those of higher working pressures, that is, from 160 to 500 lbs. per sq. in., all of the horizontal seams are butt and double strapped joints, with six rows of rivets. Each drum is provided at both ends with the Cahall patent swinging man-head. The action of this man-head is as follows: By loosening the nuts, a slight push swings the head in similar to a door, and when it is desired to again close it, it is pulled back to its place, and, owing to its being hinged, the seats come together in exactly the same place every time, thus insuring a tight joint. The flanges on the steam drums for the steam and safety valve openings are all drop forged, from flanged steel plates. Referring to Figure 3, it will be noticed that each section of tubes is connected by nipples to saddles on the steam drums. These saddles or cross boxes are made from open hearth steel, which is melted and run into molds. This steel, after cooling and annealing, presents all the chemical and physical properties of regular boiler plate steel, physical tests on a large number of coupons from these forms showing an elongation of over 25 per cent, a reduction in area of over 50 per cent, with a tensile strength of over 60,000 lbs. to the square inch.

It can be seen from the side view of the boiler (Fig. 3) that it stands on wrought iron supports and cross beams, independent of the brickwork, so that the entire structure is free to contract and expand without any strains occurring either on the setting or on the boiler itself. In this method of suspension, the entire framework is outside of the brickwork, thereby avoiding the possi-

bility of its burning away or weakening through overheating, as has frequently happened in the case of other designs.

The fronts for the boiler are what is known as the wrought iron style—that is, the entire general framework of the front is made up of wrought iron or steel beams, channels and girders, and only the panels containing the door frames are cast. This permits of a very light but rigid structure, which it is impossible to crack from the application of internal heat, which has been heretofore one of the greatest faults found with this type of boiler.

All the tubes used in this boiler are made of the best knobbled charcoal iron, which though very much more expensive than the standard iron boiler tube, yet repays the customer in future years for the additional investment in first cost. The general fittings and trimmings of the boiler are of the highest grade purchasable, the safety valves being of the solid nickel seated type. The water column used is either the Reliance or Pittsburg High and Low Water Alarm. The blow-off valves are specially made under patents owned by the Aultman & Taylor Machinery Co., and are so designed that the discs are renewable at any time, and both the disc and valve seat can be cleaned without taking the valve apart.

It will be noticed in the illustration (Fig. 4) giving the rear view of the boiler that these valves have two wheels, one directly above the other, the upper one being smaller than the lower. The larger wheel forces the disc down on its seat, the smaller wheel revolves the spindle carrying the disc. By revolving the larger wheel until the disc rests lightly on its seat and then revolving the smaller wheel, the disc is rotated on its seat, effectually

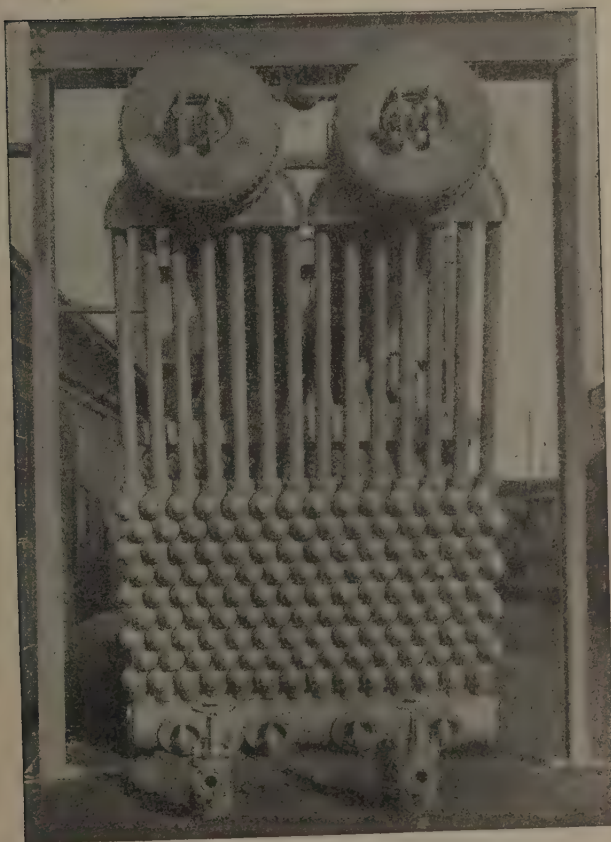


Fig. 4. Rear View of 250 H. P. Cahall Horizontal Boiler Suspended Ready for Brick Work,

clearing it of any obstructions that may have accumulated thereon.

The side cleaning doors for the boiler are of a new design, which permits the use of only one door instead of two, and when the door is opened it is thrown back entirely from the slot into which it fits, leaving a full, free opening for the introduction of the steam hose, and when the door is closed, wedge-shaped fire-brick tile, which line the door, are pushed forward in a straight line into the opening, making a perfectly smooth wall on the inside and an absolutely tight joint against the leakage of air into the setting.

Where very high pressures are to be used, say in excess of 225 pounds to the square inch, the headers or manifolds for the reception of the tubes are made of the same material used in the cross-boxes on the drums, viz., special "flowed" steel.

THE HEINE SAFETY BOILER.

Figure 5 shows a general view of the Heine water tube boiler.

The boiler is composed of the best lap welded wrought iron tubes, extending between and connecting the inside faces of two "water legs," which form the end connections between these tubes and a combined steam and water drum or "shell," placed above and parallel with them. (Boilers over 200 horsepower have two such shells.) These end chambers are of approximately rectangular shape, drawn in at top to fit the curvature of the shells. Each is composed of a *head plate* and a *tube sheet*, flanged all around and joined at bottom and sides by a butt strap of same material, strongly riveted

to both. The water legs are further stayed by *hollow stay bolts* of hydraulic tubing, of large diameter, so placed that two stays support each tube and hand hole and are subjected to only very slight strain. Being made of heavy metal, they form the strongest parts of the boiler and its natural supports. The WATER LEGS

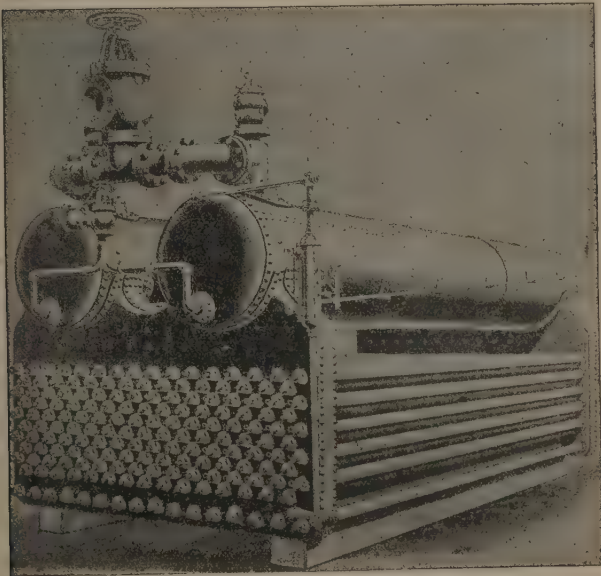


Fig 5. 375 H. P. Heine Boiler

are joined to the shell by *flanged and riveted joints* and the drum is cut away at these two points to make connection with inside of water leg, the opening thus made being strengthened by bridges and special stays, so as to preserve the original strength.

THE SHELLS are cylinders with *heads dished* to form parts of a *true sphere*. The sphere is everywhere as strong as the circle seam of the cylinder which is well known to be twice as strong as its side seam. Therefore these heads require no stays. Both the cylinder and its spherical heads are therefore *free to follow their natural lines of expansion* when put under pressure. Where flat heads have to be braced to the sides of the shell, both suffer local distortions where the feet of the braces are riveted to them, making the calculations of their strength fallacious. To the bottom of the front head a flange is riveted into which the feed pipe is screwed. This pipe is shown in the cut with *angle valve* and *check valve* attached.

On top of shell near the front end is riveted a *steam nozzle* or saddle, to which is bolted a tee. This tee carries the *steam valve* on its branch, which is made to look either to front, rear, right or left; on its top the *safety valve* is placed. The saddle has an area equal to that of stop valve and safety valve combined. The rear head carries a *blow-off flange* of about same size as the feed flange, and a *manhead* curved to fit the head, the man-hole supported by a strengthening ring outside. On each side of the shell a square bar, the *tile-bar*, rests loosely in flat hooks riveted to the shell. This bar supports the *side tiles* whose other ends rest on the *side walls*, thus closing in the furnace or flue on top. The top of the tile bar is two inches below *low water line*. The bars rise from front to rear at the rate of one inch in twelve. When the boiler is set, they must be exactly level, the whole boiler being then on an incline, i. e., with a fall of one inch in twelve from front to rear.

It will be noted that this makes the height of the

steam space in front about two-thirds the diameter of the shell, while at the rear the water occupies two-thirds of the shell, the whole contents of the drum being equally divided between steam and water.

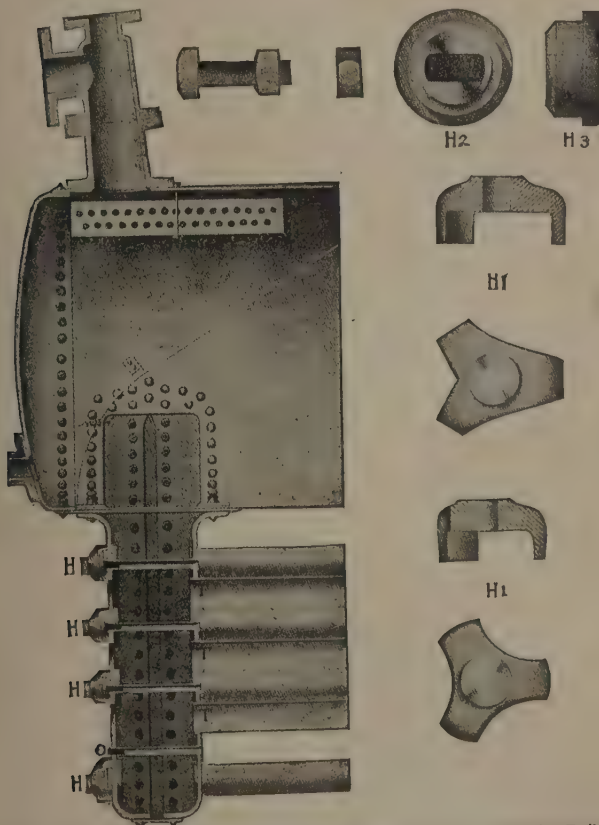


Fig. 6. Detail of Water-leg, Hand Hole Plates and Yokes, etc., of Heine Boilers.

THE TUBES extend through the tube sheets into which they are expanded with roller expanders; opposite the end of each and in the head plates is placed a hand hole of slightly larger diameter than the tube, and through which it may be withdrawn. These hand holes are closed

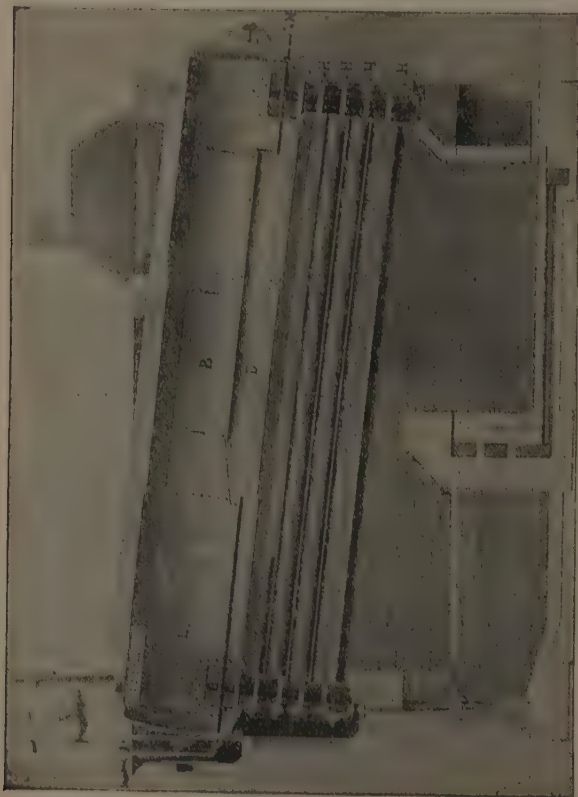


Fig. 7. Side Sectional View of Heine Boiler.

by small cast iron hand hole plates, which can be easily removed for the purpose of cleaning or inspecting a tube. Figure 6 shows these hand hole plates, marked II. In the upper corner one is shown in detail, H^2 being the top view, and H^3 a side view, while H^1 is the yoke or crab placed outside to support the bolt and nut. Figure 7 is a longitudinal section showing inside construction. The mud drum D is located well below the water line, parallel to and three inches above the bottom of the shell. It is of oval section, slightly smaller than the manhole. It is entirely enclosed, except about eighteen inches of its upper portion at the front end, which is cut away nearly parallel with the water line. The mud drum D is made of strong sheet iron, with cast iron heads, and its action is as follows:

The feed water enters it through the pipe F about one-half inch above its bottom; even if it has previously passed the best heaters it is colder than the water in the boiler. Hence it drops to the bottom, and, impelled by the pump or injector, passes at a *greatly reduced speed* to the rear of the mud drum. As it is gradually heated to near boiler temperature it rises and flows slowly in reverse direction to the open front of the mud drum; here it passes over in a *thin sheet* and is immediately swept backward into the main body of water by the swift circulation, thus becoming *thoroughly mixed* with it before it reaches the tubes. During this process the mud, lime, salts, and other precipitates are deposited as a sort of semi-fluid "sludge" near the rear end of the mud drum, whence it is blown off at frequent intervals through the blow-off valve N. As the speed in the mud drum is only about one-fiftieth of that in the feed water pipe, plenty of time is given for this action. Any

precipitates which may escape the mud drum at first, will of course form a scale on the inside of the tubes, etc. But the action of expansion and contraction cracks off scale on the *inside* of a tube *much faster* than on the *outside*, and then the circulation sweeps the small chips, like broken egg-shells, upward, and as they pass over the mouth of the mud drum *they drop in the eddy*, lose velocity in this slow current and fall to the bottom, and, being pushed by the feed current to the rear end, are blown off from the mud drum with other refuse. On opening a Heine boiler after some months' service, such bits of scale, whose shape identifies them, are always found in the mud of the mud drum. *Very little loose scale* is found on the bottom of the water legs; the current through the lower tubes, always the swiftest, brushes too near the bottom to allow much to lodge there.

This explanation of the action of the mud drum shows how *the inside of the tubes may be kept clean*. To keep the *outside clear of soot and ashes* which deposit on, and sometimes even bake fast to the tubes, each boiler is provided with two special nozzles, with both side and front outlets, a short one for the rear, a long one for the front. They are of three-eighth inch gas pipe and each is supplied with steam by a one-half inch steam hose. The nozzle is passed through each stay bolt in turn, and thus delivers its side jets on the three or four tubes adjacent, with the full force of the steam, at the short range of two inches, *knocking the soot and ashes* off completely, while the end jet carries them into the main draft current, to lodge at points in breeching or chimney base convenient for their ultimate removal. An inspection of the cuts will show that the stay bolts are so located that the nozzle can in turn be brought to

bear on all sides of the tubes. As soon as the nozzle is withdrawn from the stay bolt this is closed air-tight by a plain wooden plug.

In cleaning a boiler it is only necessary to *remove every fourth or fifth handhole plate* in the front water leg; the water hose, supplied with a short nozzle, can be entered in all the adjacent tubes, owing to the ample dimensions of the water leg. In the rear water leg only one or two handholes in the lower row need be opened to let the water and debris escape. The others in rear water leg are frequently *left untouched for years*. A lamp or candle hung on a wire through the manhead may be held opposite each tube so that it can be perfectly inspected from the front.

The feed pipe F enters the mud drum through a loose joint in front, and the blow-off pipe N is screwed tightly into its rear head, passing by a steam-tight joint through the rear head of the shell. Just under the steam nozzle is placed a dry pipe A. L is a deflection plate, extending from the front of the shell, and inclined upwards, beyond the mouth of the front water leg. The throat or mouth of each water leg is large, to equal in area the total tube area, and where it joins the shell it increases gradually in width by double the radius of the flange. In the setting, the front water leg is placed firmly on a set of strong cast iron columns, bolted and braced together by the door frames, dead plate, etc., forming the fire front. This is the fixed end. The rear water leg rests on rollers, free to move on cast iron plates set in the lower masonry of the rear wall. The brick work does not close in entirely to the boiler, the space between being filled with tow or waste saturated with fire clay or other refractory, but pliable material, thus leaving

the boiler and its walls each free to move separately during expansion or contraction, without disturbing any joints in the masonry. On the lower, and between the upper tubes, are placed light fire brick tiles. The lower tier extends from the front water leg to within a few feet of the rear one, leaving there an upward passage across the rear ends of the tubes for the flame and gases. The upper tier closes into the rear water leg, and extends forward to within a few feet of the front one, thus leaving an opening for the gases in front. The side tiles extend from the side walls to tile bars, and close up to the front water leg, and front wall, and leave open the final uptake for the waste gases over the back part of the shell, which is here covered above the water line with a row of lock fire brick, resting on the tile bars. The rear wall, and one parallel to it, are arched over the shell a few feet forward, form the uptakes.

THE BABCOCK & WILCOX WATER TUBE BOILER.

Description. Figure 8 presents a side view of the Babcock & Wilcox boiler, and Figure 9 a partial section.

This boiler is composed of lap-welded wrought iron tubes, placed in an inclined position and connected with each other, and with a horizontal steam and water drum, by vertical passages at each end, while a mud-drum is connected to the rear and lowest point in the boiler.

The end connections are in one piece for each vertical row of tubes, and are of such form that the tubes are "staggered" (or so placed that each row comes over the spaces in the previous row). The holes are accurately sized, made tapering, and the tubes fixed therein by an expander. The sections thus formed are connected with

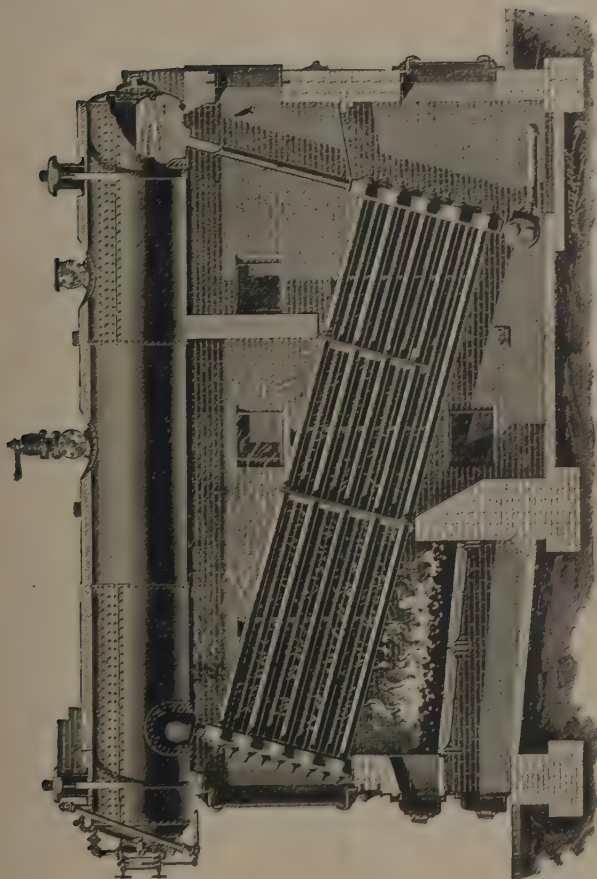


Fig. 8. Side View of Babcock & Wilcox Boiler of Wrought Steel Construction.

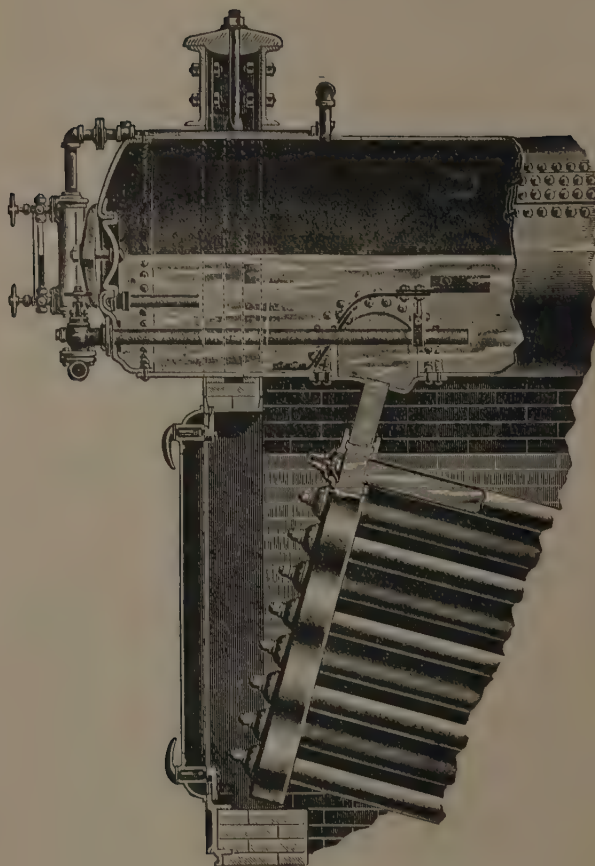


Fig. 9. Partial Vertical Section Babcock & Wilcox Boiler.

the drum, and with the mud-drum also by short tubes expanded into bored holes, doing away with all bolts, and leaving a clear passageway between the several parts. The openings for cleaning opposite the end of each tube are closed by hand-hole plates, the joints of which are made in the most thorough manner, by milling the surfaces to accurate metallic contact, and are held in place by wrought-iron forged clamps and bolts. They are tested and made tight under a hydrostatic pressure of 300 pounds per square inch, *iron to iron, and without rubber packing or other perishable substances.*

The steam and water drums are made of flange iron or steel, of extra thickness, and double riveted. They can be made for any desired pressure, and are always tested at 50 per cent above the pressure for which they are constructed. The mud-drums are of cast iron, as the best material to withstand corrosion, and are provided with ample means for cleaning.

Erection. In erecting this boiler, it is suspended entirely independent of the brickwork, from wrought iron girders resting on iron columns. This avoids any straining of the boiler from unequal expansion between it and its enclosing walls, and permits the brickwork to be repaired or removed, if necessary, without in any way disturbing the boiler. All the fixtures are extra heavy and of neat designs.

Operation. The fire is made under the front and higher end of the tubes, and the products of the combustion pass up between the tubes into a combustion chamber under the steam and water drum; from thence they pass down between the tubes, then once more up through the spaces between the tubes, and off to the chimney. The water inside the tubes, as it is heated, tends to rise



Fig. 10. End View of Hender,
Babcock & Wilcox Boiler.



Forged Steel Cross Box.

Fig. 11. Babcock & Wilcox Boiler.



Forged Steel Header Hand Hole Fittings.

Fig. 12. Babcock & Wilcox Boiler.



Forged Steel Drum Head.

Fig. 13. Babcock & Wilcox Boiler.

towards the higher end, and as it is converted into steam—the mingled column of steam and water being of less specific gravity than the solid water at the back end of

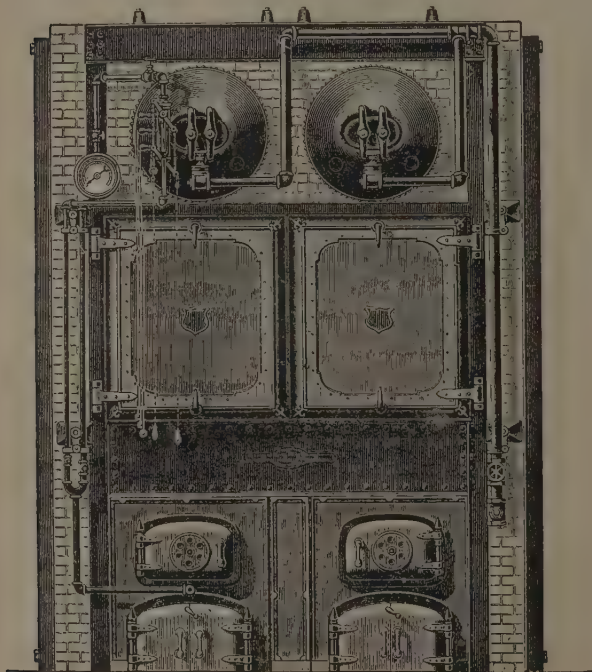


Fig. 14. Standard Front of Babcock & Wilcox Boiler.

the boiler—rises through the vertical passages into the drum above the tubes, where the steam separates from the water and the latter flows back to the rear and down again through the tubes in a continuous circulation. As

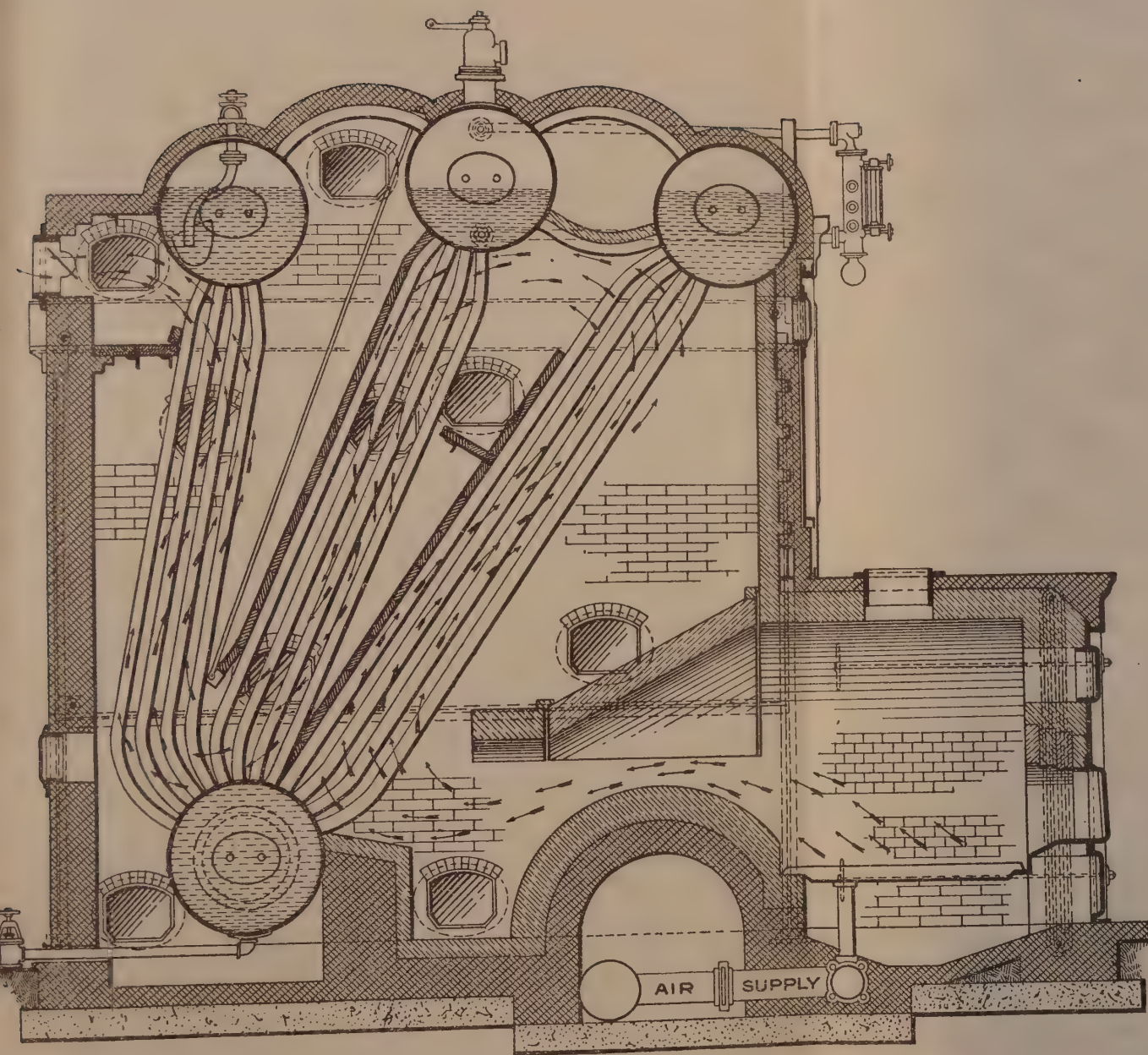


FIG. 15

SECTIONAL SIDE AND FRONT ELEVATION OF STIRLING BOILER AND BAGASSE FURNACE

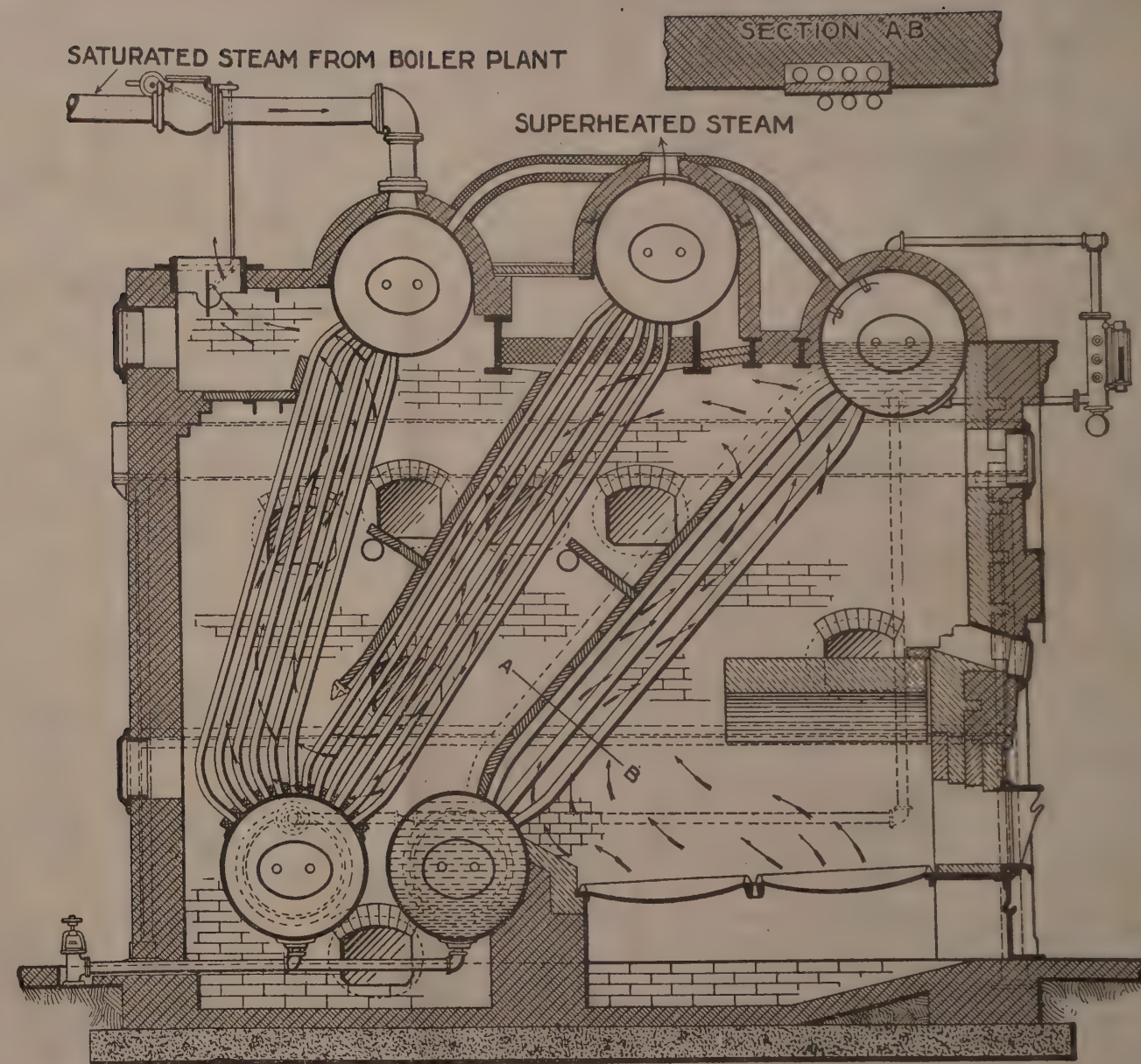
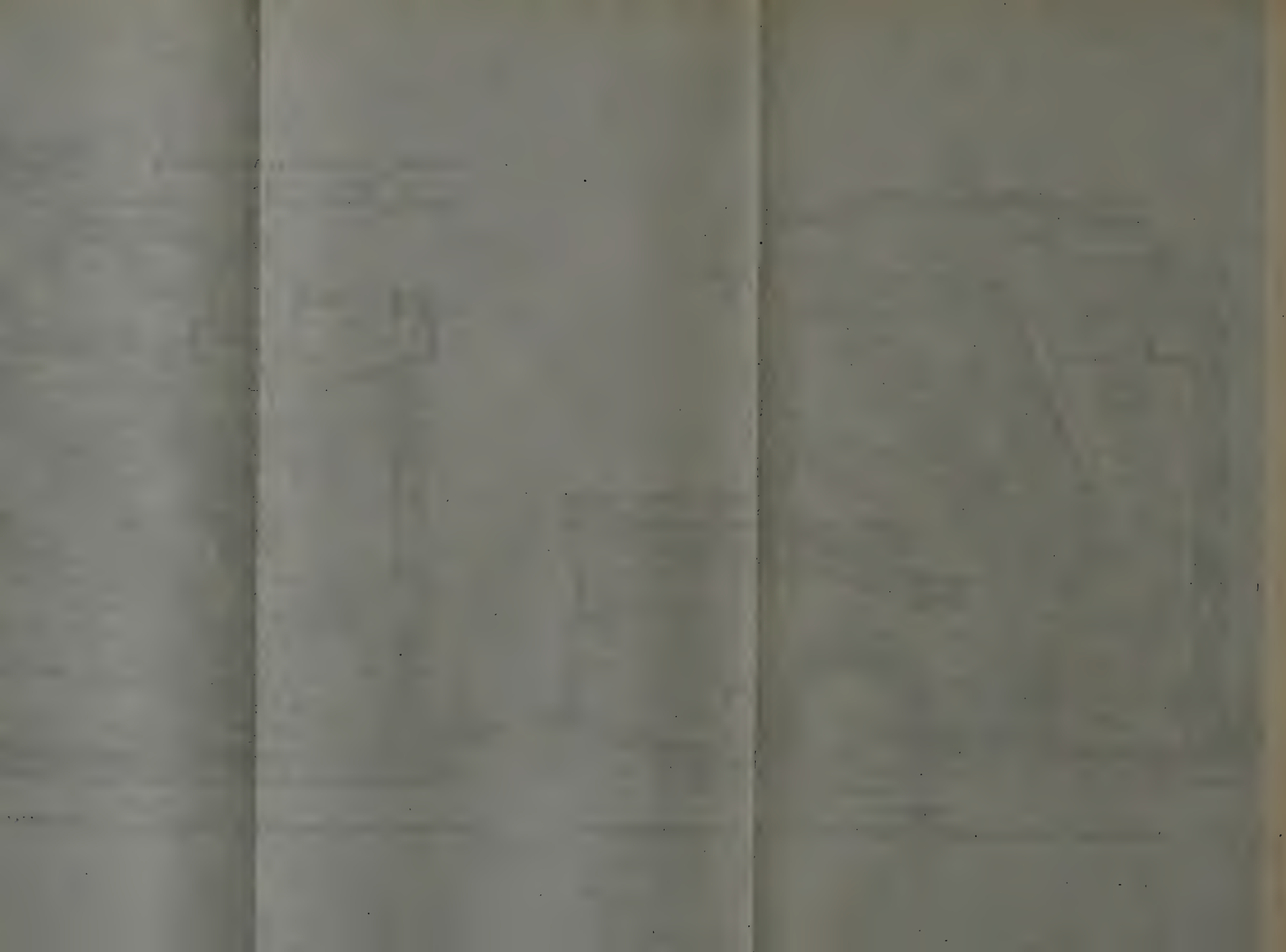


FIG. 16

SECTIONAL SIDE ELEVATION OF THE STIRLING INDEPENDENTLY FIRED SUPERHEATER



the passages are all large and free, this circulation is very rapid, sweeping away the steam as fast as formed, and supplying its place with water; absorbing the heat of the fire to the best advantage; causing a thorough comingling of the water throughout the boiler and a consequent equal temperature, and preventing, to a great degree, the formation of deposits or incrustations upon the heating surfaces, sweeping them away and depositing them in the mud-drum, whence they are blown out.

The steam is taken out at the top of the steam-drum near the back end of the boiler after it has thoroughly separated from the water, and to insure dry steam, a perforated dry-pipe is connected to the nozzle inside the drum.

THE STIRLING WATER-TUBE SAFETY BOILER.

The Stirling boiler, figures 15 and 16, consists of three upper or steam drums, each connected by a number of tubes (called a "bank") to a lower or mud drum. Suitably disposed fire tile baffles between the banks direct the gases into their proper course. Shorter tubes connect the steam spaces of all upper drums, also water spaces of front and middle drums. The boiler is supported on a structural steel framework, around which is built a brick setting, whose only office is to provide furnace space, and serve as a housing to confine the heat. The entire front is of metal of appropriate and artistic design. These parts, together with the usual valves and fittings, constitute the completed boiler, which represents the acme of simplicity and eliminates the complication of the older types.

The drums vary from 36 to 54 inches in diameter and

are made of the best open hearth flange steel. The plates extend the entire distance between heads, hence there are no *circular seams*. The longitudinal seams—which are double or triple riveted according to the working pressure to be carried—are so placed that they are not exposed to high temperature. The drum heads are hydraulically dished to proper radius; each drum is provided with one manhole, and the manhole plate and arch bars are of wrought steel; four manhole plates, which can be removed in ten minutes, give access to the entire interior of the boiler, and expose every tube end, rivet and joint. The drum interiors are perfectly clear; there are no baffles, stays, tie-rods, mud pipes or other obstructions in them.

The tubes are best lap-welded mild steel. They are slightly curved at the ends to permit them to enter the drums normally and to provide for free expansion of the boiler when at work. The tubes are expanded directly into reamed holes in tube sheets of the drums, hence the annular recess between tubes and the cast headers of some types of boiler is eliminated, and failure of tubes by pitting through corrosion, caused by accumulation of soot in these recesses, is avoided. There are no short nipples and no tube joints in places which can be reached only by jointed handles on the tube expander, rendering it impossible to determine when the tube has been properly expanded. In the Stirling boiler every tube end is visible and accessible. As the entire weight of boiler and contents is supported on the steel frame work, cracking of the setting, due to unequal settlements, is obviated, and no blocking is needed when the brick work has to be repaired. The design of frame work can be modified to suit special conditions. The brick

setting is so clearly shown in figures 15 and 16 that an extended description is unnecessary. No special shapes or other material not found in open market are needed. Any necessary repairs to the brick work can be made without disturbing the boiler connections. In the design of the Stirling furnace it will be seen by reference to figures 15 and 16 that a fire brick arch is sprung over the grates, and immediately in front of the first bank of tubes. The large triangular space between boiler front, tubes and mud drum is available for a combustion chamber, and for installation of sufficient grate surface to meet the requirements of the lowest grades of fuel.

Baffles and Course of Gases. The baffle walls rest directly upon the tubes, and guide the course of the gases up the front bank, down the middle and up the rear bank, thus bringing them into such intimate contact with the boiler surface that the heat is quickly and thoroughly extracted from them. In no other boiler are the gases compelled to travel as far before reaching the stack, and the effect upon economy is evident. The baffles are made of plain rectangular firetile carried in stock by all fire-brick dealers, in contrast to the special formed bricks (obtainable only from the manufacturer) required by many types of water-tube boiler. Another marked advantage of the Stirling baffles is that since no tubes pass *between* or through the tiles (see Fig. 17), they are not pried apart and made leaky by distorted tubes; they can be removed and replaced without disturbing a tube. Baffles built across the tubes, as in many boilers, are damaged by pulling a faulty tube through them, and can be repaired in but one way—by removal of every tube necessary to permit a man to crawl in and reach the defective spot.

Simplicity. There are no details of complicated shape; no flat surfaces, tie-rods, water-legs, headers, return-bends, outside circulating pipes to plug up; no multitudinous handhole plates to be removed and packed with gaskets, or be ground and scraped to a fit whenever boiler is opened; no baffles or mud pipes in the drums; no short nipples, seams exposed to heat, or parts inaccessible for cleaning.

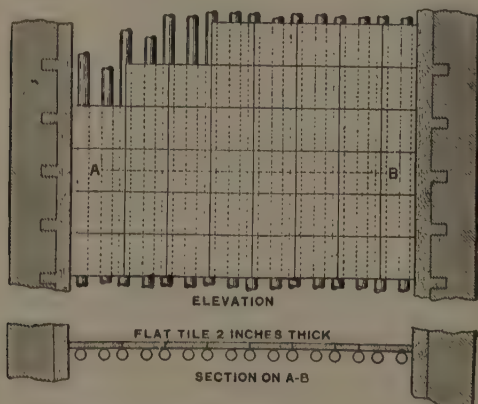


Fig. 17. Elevation and Section of Firetile Baffles in Stirling Boilers.

Expansion and Contraction. In the Stirling the mud drum is not embedded in brickwork, but is suspended on the tubes which connect it with the upper drums.

In consequence of this construction, not only may the mud drum with perfect freedom move an amount representing the resultant expansion of the boiler, but any difference in expansion between the individual tubes, such as caused when one side of the furnace is being

cleaned and other side is excessively hot, is taken up by the curve in the tube. The boiler therefore stays tight, and is entirely free from the stresses and frequent leaks caused by unequal expansion of straight tubes rigidly connected at each end to headers, water-legs, or large drums. It will thus be seen that the bent tube performs in the boiler the same function as an expansion loop in a steam line, and that its successful introduction in the Stirling boiler is a distinct and far-reaching advance in steam engineering.



Fig. 18. Stirling Boiler Cleaning Doors, with Asbestos Packing Washers.

Rapid Circulation. The path of the circulation in the Stirling is as follows: The water is fed into upper rear drum, passes down the rear bank of tubes to the lower drum, thence up the front bank to forward steam drum. Here the steam formed during passage up the front bank disengages and passes through the upper row of cross tubes into the middle drum, while the *solid water* passes through the lower cross tubes into middle drum, then down the middle bank to lower drum, from which it is again drawn up the front bank to retrace its former course until it is finally evaporated. The steam generated in the rear bank passes through cross tubes to the center drum.

The temperature of gases in contact with the tubes will

evidently be greatest at the bottom of the front bank, and gradually decreases as the gases proceed along their course to the breeching. Obviously then the velocity of water circulation and quantity of steam generated will be a maximum in the front bank; in the rear bank there is a slow circulation downward equal to the quantity of water evaporated in the other two banks. The peculiar benefits arising from this action will be discussed under caption, "Handling Impure Feed Water."

Rapid circulation is essential for the following reasons:

(1) To keep all parts of the boiler at practically the same temperature, thus eliminating severe stresses due to unequal expansion.

(2) To permit quick raising of steam and rapid response to sudden demands on the boiler capacity.

(3) To sweep away from the heating surfaces all steam bubbles as fast as formed, and thereby prevent "steam pockets," which quickly burn out the tubes. This is so particularly the case where intense local heating occurs due to use of gas or oil fuel, that some types of boiler fairly well adapted to coal cannot be successfully used with these fuels.

The third requirement is met only indifferently or not at all in those types of boilers in which tubes often numbering as many as eighteen must discharge their entire content of steam and water through a narrow water-leg, or, worse still, through a single nipple whose cross section is equal to that of but one tube. At 150 pounds' gauge one cubic foot of water, when converted into steam, will have a volume of about 151 cubic feet. In consequence of this great increase in volume, as soon as the boiler is forced the nipple area becomes insufficient, steam pockets form in the lower tubes, which then be-

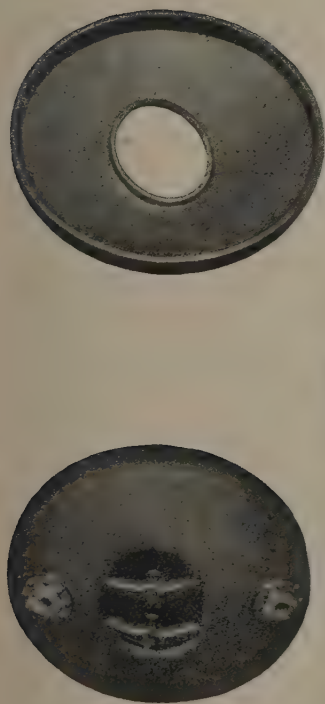
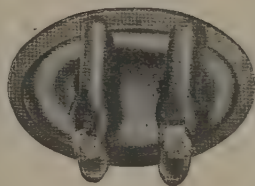
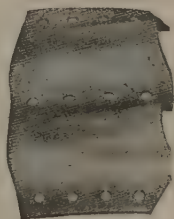


Fig. 19. Stirling Boiler Forged Steel Drum Heads and Pads for Water Column Connections.



Manhole and Arch Bars.



Drum Lug.



Steam Nozzle Pad.

Fig. 20. Forged Steel Details of the Stirling Boiler.

come overheated, and buckle and leak, and finally burn out. So inadequate are these nipples and headers that recent experiments of M. Brull have shown that in boilers whose circulation is constricted by nipples or narrow water-legs, the circulation in the upper tubes *reverses*, that is, it goes from the front to rear instead of in the opposite way, as intended. In consequence of this, much matter suspended in the water is swept into the bottom tubes, which fact, in connection with the steam pockets, explains why those tubes so rapidly fail.

In the Stirling boiler there is *no constriction of the circulation*, as each tube discharges directly into the drums, without intervention of headers, nipples or water-legs. The nearly vertical position of the tubes also promotes rapid circulation, hence steam pockets cannot form, and a fruitful cause of interrupted service and tube renewals is thus eliminated.

A most fruitful cause of burnt tubes is a piece of scale which becomes detached and falls on the bottom of the tube, and the spot under it is certain to burn out quickly. The Stirling is free from this source of tube destruction, because while the scale will not form in the hotter tubes unless the boiler is neglected, even if it does form owing to such neglect and a piece becomes detached, it will slide down to the mud drum instead of lodging.

Cleaning the Interior. By removing four manhole plates, which can be done in ten minutes, the entire boiler interior is accessible for cleaning. From the preceding discussion it is evident that the precipitates are settled into the mud drum, whence they are blown off at intervals; the scale is practically confined to the rear bank of tubes, and by reason of escaping the high temperatures it is soft and easily detached. Consequently it happens

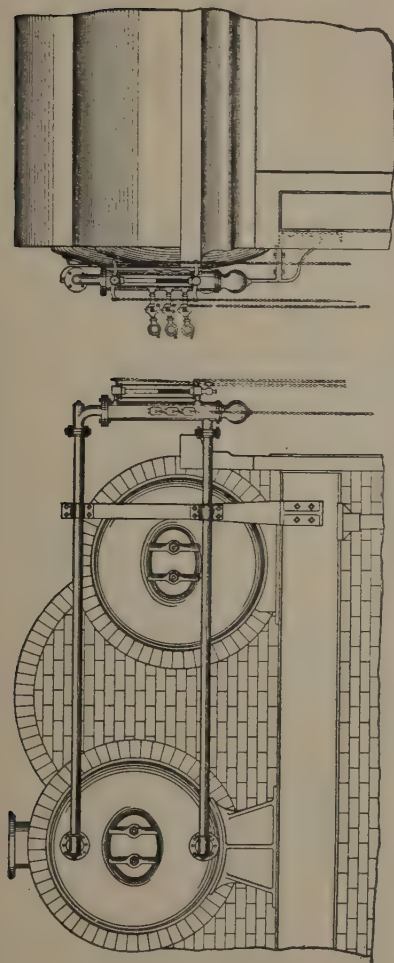


Fig. 21. Stirling Boiler Water Column and Connections, with Quick-closing Gauge Glass Fittings.

in most cases that only the rear bank needs cleaning each time the boiler is opened, while the others need only occasional attention. The scale is quickly and cheaply removed by a "turbine cleaner," described and illustrated in the section on Boiler Operation.

Cleaning the Exterior. Ample cleaning doors are provided both in the sides and rear of the setting, so that the exterior of the heating surfaces may be kept clean and all accumulations of soot, ashes, etc., blown off as rapidly as they form, by using a steam blower-pipe which is furnished with every boiler.

Durability. By reason of the elimination of thick plates and riveted joints exposed to the fire; cast metal of all kinds; parts of irregular shape and uncertain strength; stresses due to unequal expansion, multitudes of caps, joints and nipples, and similar objectionable details, the Stirling boiler is free from parts liable to get out of order. The prevention of scale deposits in the hottest tubes; the perfect facilities for keeping the boiler clean; the rapidity of water circulation and impossibility of forming steam pockets, all combine to protect the tubes against burning out. Hence the necessity of repairs to the boiler itself is extremely remote. The setting is simple and substantial and not subject to derangement other than the natural wear of surface lining.

"MAXIM" WATER TUBE BOILER.

The "Maxim" water-tube boiler (Figures 22 and 23) consists of two drums, one above the other, connected by tubes. Each tube has two bends, thus providing for unequal expansion and contraction. The space between the tubes is greater than the diameter of the tubes, so

that any tube can be passed through the space between. The tubes, which are nearly vertical, are arranged in parallel rows, the lower tube plates being cylindrical,

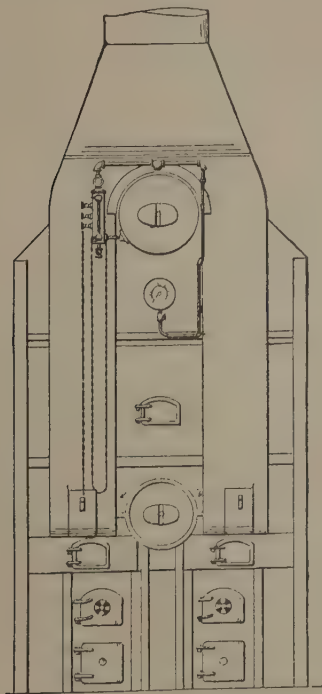


Fig. 22. Front Elevation Maxim Boiler.

and the bottom end of every tube is opened to the lower drum, thus enabling the flue dust, mud and loose scale to drop away from the heating surface. There are no

riveted joints exposed to the flames or fire gases, and there are no cylinders subject to external pressure.

The heating surface is arranged so as to break up the current of heated gases, which travels three times the length of the tube, the form and direction of the gas current being changed seven times between the furnace and chimney, to insure a low temperature to the escaping gases.

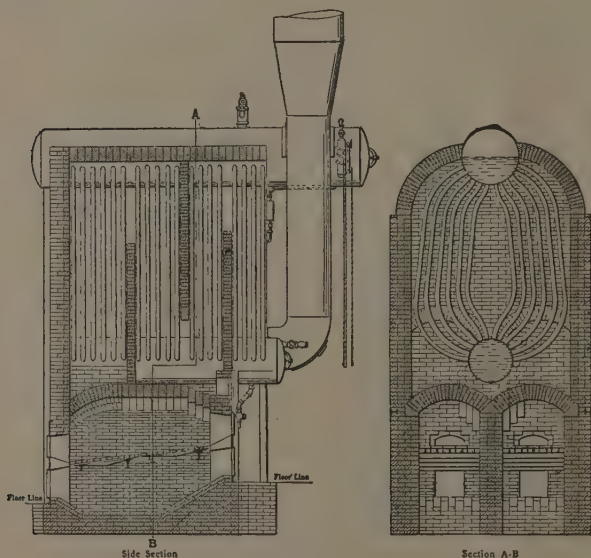


Fig. 23. Maxim Boiler.

The circulation is constant, the return circulation being provided for by a third set of tubes, which are subjected to the heat of the gases just before reaching the chimney. As the cold feed water meets the gases as they leave

the boiler, the front section of the tube, which receives the feed water, acts as an economizer and purifier

The furnace of this type of boiler is constructed of fire-brick, and is built under the boiler. Between the furnace and each combustion chamber there is a throat contracted to the proper size for drafts, the purpose being to insure a better combustion by an intimate mixture of gases. The illustrations show a side view, sectional view through *A B*, and a front view.

THE BIGELOW-HORNSBY WATER-TUBE BOILER.

The American rights to manufacture the Hornsby water-tube boiler, which has been on the English market five years, were recently acquired by the Bigelow Company, of New Haven, Conn., hence in this country it will be known as the "Bigelow-Hornsby" boiler. It has been re-designed in a measure, to meet the requirements of American high-pressure practice. A general idea of its construction may be obtained from Fig. 23a, which shows a section through the setting. It will be noted that the boiler is supported entirely from the overhead beams, leaving the lower part free to respond to expansion stresses. The front tubes are inclined at an angle of 68 degrees, and the rear units are vertical, as shown. Only two lengths of tubes are used. Each section is independent of its neighbor, except the nipples connecting with the steam drum and the equalizing nipples which connect the bottom drums of the rear sections. This flexible form of construction permits the building of very large units, even larger than the 2,200-horse-power Hornsby boilers at the Bow Street station of the Charing Cross & London Electricity Works, where four

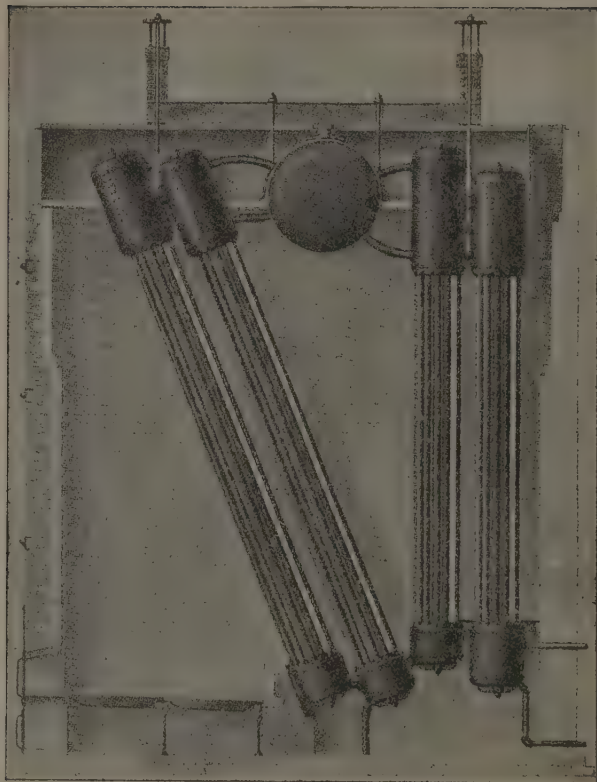


Fig. 23a. Section Through Setting of Bigelow-Hornsby Boiler.

boilers containing 21,700 square feet of heating surface evaporate in regular service 110,000 pounds of water per hour. A large percentage of the heating surface is exposed to the radiant heat of the furnace, and to the first

pass of gases, before these have reached any other heating surface. The tubes of the front unit, which are located in front of the baffles, comprise more than 12 per cent of the heating surface of the boiler.

The feed-water is admitted into the bottom drum of the rear unit and is advanced gradually from the coldest to the hottest portion of the boiler, first passing up the

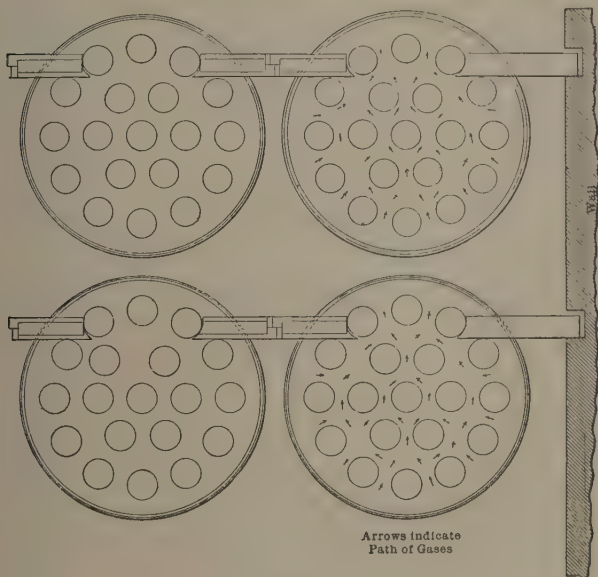


Fig. 23b.

entire length of the tubes in the rear unit, then from the top of those tubes to the unit immediately in front, down this unit, and up the two front units and back through the steam drum to the first vertical unit at the

rear of the steam drum. There is also, as may be conceived, a rapid local circulation in each of the units while the general circulation is going forward. The speed of the feed-water up the rear unit being regulated by the amount of steam generated, ample time is permitted for sediment and scale-forming matter to be deposited in the bottom drum of this unit. All liberation of steam from the water surfaces takes place in the upper drums and is entirely unrestrained, the full area of the tube openings communicating with the drums. The transfer of steam and water between units occurs through separate nipples, and the water nipples are required merely to care for the general circulation through the boiler.

The arrangement of this boiler is such that it can be baffled so that the products of combustion are carried uniformly over the heating surfaces in thin layers, there being no large unrestricted paths parallel to the heating surfaces through which the gases can flow in their passage to the uptake. Fig. 23b, which is a horizontal section through some of the rear units, shows how they are arranged with reference to the gas currents, and how the baffle-plates serve to guide the gases through in substantially uniform passages.

The smoke flue can be taken off at the back of the boiler at any point between the top and bottom. All the tubes are straight and every tube in each section can be reached for cleaning by the removal of a single manhole cover.

The boiler is built to a factor of safety of five for 200 pounds working pressure, and it is stated that a test section has been subjected to hydrostatic pressure of 1,000 pounds without rupture. The ratio of grate surface to heating can be made as low as 1 to 26.

QUESTIONS AND ANSWERS.

1. What types of boilers are most commonly used in central power stations supplying electric current for railway service?

Ans. The horizontal tubular boiler and the water tube boiler.

2. Describe in general terms the horizontal tubular boiler.

Ans. It consists of a shell having tubes of small diameter, extending from head to head.

These tubes are located in the water space.

3. What is their function?

Ans. To supply a passageway to the stack for the hot gases from the furnace.

4. Does the water in the boiler receive heat from these tubes?

Ans. It certainly does.

5. Describe the route taken by the smoke and hot gases in the operation of a tubular boiler.

Ans. From the furnace, located under the front end of the boiler, the gases pass under and along the sides of the shell, back to the rear end, the upper part of which is arched over. The route is here reversed, and the products of combustion return through the flues towards the front end and thence through the breeching into the stack.

6. Is this type of boiler economical in the burning of fuel?

Ans. It can be made so if properly set and handled in operation.

7. Describe in a general way the water tube boiler.

Ans. It consists of a set, or sets of tubes 3 to 4 in. in diameter, sometimes vertical, and sometimes inclined, and connected at the top to a steam drum, and at the bottom to a mud drum.

8. What advantages as regards circulation of the water has the water tube boiler?

Ans. It provides for a free circulation.

9. Name another advantage connected with the water tube boiler.

Ans. The margin of safety from dangerous explosions.

10. Why is this?

Ans. Because if one or more tubes give way the pressure is relieved.

11. What precautions should be observed in the design and construction of a boiler?

Ans. The best materials should be used, the boiler should be simple in design, and the workmanship should be perfect.

12. Where should the mud drum be located?

Ans. In a place removed from the action of the fire.

13. What should be the capacity of the boiler relative to its work?

Ans. It should have a steam and water capacity sufficient to prevent any fluctuation in either the steam pressure or the water level, if properly fed.

14. Why should the water in a boiler circulate freely and constantly?

Ans. In order to maintain all parts at as near the same temperature as possible.

15. What should the strength of a boiler be, relative to the strain it is liable to be subject to?

Ans. It should have a great excess of strength.

16. Is a combustion chamber an advantage to a boiler?

Ans. It is, in order to complete the combustion of the gases before they escape to the chimney.

17. How should a boiler be arranged with regard to cleaning?

Ans. All parts should be easily accessible for cleaning and repairs.

18. What type of boiler is the Cahall?

Ans. It is a water tube boiler.

19. Is it vertical or horizontal?

Ans. It is built either way.

20. What form of Cahall is generally used in central power stations?

Ans. The horizontal form.

20a. What is the range of pressures that these boilers are built for?

Ans. From 160 to 500 lbs. per sq. in.

21. Describe the method of constructing the joints.

Ans. The sheets are beveled on the edges, bent into shape, and rivet holes drilled after bending.

22. What is gained by so doing?

Ans. Absolutely round rivet holes and no crystallization.

23. What type of riveted joint is used on the higher pressure boilers?

Ans. Triple riveted, double strapped.

24. How are the tubes connected to the steam drum in the Cahall boiler?

Ans. By nipples connected to saddles on the drum.

25. Does this boiler rest upon the brick work?

Ans. It does not, but is suspended free from the masonry.

26. What advantage is there in this style of setting?

Ans. The entire structure is free to expand or contract without causing any strains on either boiler or brick work.

27. Describe the Heine boiler.

Ans. It consists of one, and sometimes two shells on drums resting upon water legs riveted to each end. These water legs are connected by horizontal tubes. The water fills the tubes, water legs, and partially fills the shell, leaving the upper portion for steam space.

28. In the setting does this boiler occupy a horizontal position?

Ans. No. The shell and tubes have an incline of one inch in twelve from front to rear.

29. What provision is made for cleaning and repairing the tubes?

Ans. Hand holes are located in the head plates opposite each tube.

30. How are these hand holes closed?

Ans. In the ordinary way, by plates.

31. Where is the mud drum located in the Heine boiler?

Ans. Inside the shell, near the bottom.

32. How is the Heine boiler supported in the setting?

Ans. The front or fixed end rests upon cast iron columns. The rear water leg upon rollers.

33. Describe in brief the Babcock and Wilcox boiler.

Ans. It is composed of wrought iron tubes, placed in an inclined position, and connected with each other, and

with a horizontal steam and water drum by vertical headers.

34. Where is the mud drum in this boiler?

Ans. In the rear, and connected to the lowest part of the boiler.

35. What provision is made for cleaning the tubes in the Babcock and Wilcox boiler?

Ans. Through hand holes in the headers, opposite each tube?

36. How is this boiler supported in the setting?

Ans. It is suspended from wrought iron girders, entirely independent of the brick work.

37. Describe in general terms the Stirling boiler.

Ans. It consists of three upper steam drums, each being connected by a number of tubes to a lower or mud drum.

38. How are the steam spaces connected?

Ans. By shorter tubes.

39. How is the boiler supported?

Ans. On a structural steel frame work.

40. What provision is made for expansion and contraction of the tubes?

Ans. They are slightly curved near the ends.

41. How are the hot gases directed in their course from furnace to stack?

Ans. By means of fire brick baffle walls.

42. How is the interior of this boiler cleaned?

Ans. Four manholes are provided in the drums, by which access to the interior of both the drums and tubes is obtained.

43. What type of boiler is the "Maxim" boiler?

Ans. It is a water tube boiler consisting of two drums, one above the other, connected by tubes.

44. Describe the tubes?

Ans. Each tube has two bends, thus providing for unequal expansion or contraction.

45. How is the heating surface of the "Maxim" boiler arranged?

Ans. It is so arranged that the current of heated gases is made to travel three times the length of the tubes, the direction of the current being changed seven times in its route from furnace to stack.

BOILER CONSTRUCTION.

As it is of the highest importance not only to the engineer in charge of the plant, but also to his assistants, and in fact to all persons whose business compels them to be in the vicinity of the boiler-room, that there should be absolutely no doubt as to the safe construction of the boilers and their ability to withstand the pressures under which they are operated, the author has compiled the following data regarding the construction and strength of boilers. The deductions and reports of tests and experiments made by such eminent authorities as Dr. Thurston, Prof. Wm. Kent, Dr. Peabody, D. K. Clark, Hutton, and many other experts have been consulted, and the author has also added the results of his own observations, collected during an experience of thirty-five years as a practical engineer.

When steel was first introduced as a material for boiler plate, it was customary to demand a high tensile strength, 70,000 to 74,000 lbs. per sq. in., but experience and practice demonstrated in course of time that it was much safer to use a material of lower tensile strength. It was found that with steel boiler plate of high tenacity there was great liability of its cracking, and also of certain changes occurring in its physical properties, brought about by the variations in temperature to which it was exposed. Consequently present-day specifications for steel boiler plate call for tensile strengths running from 55,000 to 66,000 lbs., usually 60,000 lbs. per sq. in. Dr. Thurston gives what he calls "good specifications" for

boiler steel as follows: "Sheets to be of uniform thickness, smooth finish, and sheared closely to size ordered." Tensile strength to be 60,000 lbs. per sq. in. for fire box sheets and 55,000 lbs. per sq. in. for shell sheets. Working test: a piece from each sheet to be heated to a dark cherry red, plunged into water at 60° and bent double, cold, under the hammer. Such piece to show no flaw after bending. The U. S. Board of Supervising Inspectors of Steam Vessels prescribes, in Section 3 of General Rules and Regulations, the following method for ascertaining the tensile strength of steel plate for

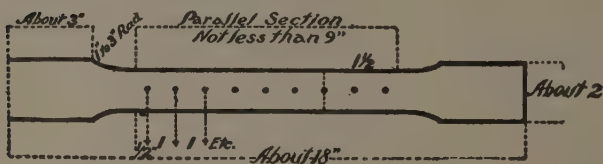


Fig. 24. Test Piece.

boilers: "There shall be taken from each sheet to be used in shell or other parts of boiler which are subject to tensile strain, a test piece prepared in form according to figure 24. The straight part in center shall be 9 in. in length and 1 in. in width marked with light prick punch marks at distances 1 in. apart, as shown, spaced so as to give 8 in. in length. The sample must show, when tested, an elongation of at least 25 per cent in a length of 2 in. for thickness up to $\frac{1}{4}$ in., inclusive; in a length of 4 in., for over $\frac{1}{4}$ in. to $\frac{7}{16}$ in., inclusive; in a length of 6 in., for all plates over $\frac{7}{16}$ in. and under $1\frac{3}{4}$ in. in thickness. The samples shall also

be capable of being bent to a curve of which the inner radius is not greater than $1\frac{1}{2}$ times the thickness of the plates, after having been heated uniformly to a low cherry red and quenched in water of 82° F.

Punched and Drilled Plates. Much has been written on this subject, and it is still open for discussion. If the material is a good, soft steel, punched sheets are apparently as strong and in some instances stronger than drilled, especially is this the case with regard to the shearing resistance of the rivets, which is greater with punched than with drilled holes.

Concerning rivets and rivet iron and steel, Dr. Thurston has this to say in his "Manual of Steam Boilers": "Rivet iron should have a tenacity in the bar approaching 60,000 lbs. per sq. in., and should be as ductile as the very best boiler plate when cold. A good $\frac{5}{8}$ -in. iron rivet can be doubled up and hammered together cold without exhibiting a trace of fracture." The shearing resistance of iron rivets is about 85 per cent and that of steel rivets about 77 per cent of the tenacity of the original bar, as shown by experiments made by Greig and Eyth. The researches made by Wöhler demonstrated that the shearing strength of iron was about four-fifths of the tensile strength.

The tables that follow have been compiled from the highest authorities and show the results of a long and exhaustive series of tests and experiments made in order to ascertain the proportions of riveted joints that will give the highest efficiencies.

The following Table 1 gives the diameters of rivets for various thicknesses of plates and is calculated according to a rule given by Unwin:

TABLE 1.

TABLE OF DIAMETERS OF RIVETS.

Thickness of Plate.	Diameter of Rivet.	Thickness of Plate.	Diameter of Rivet.
$\frac{1}{4}$ inch	$\frac{1}{2}$ inch	$\frac{9}{16}$ inch	$\frac{7}{8}$ inch
$\frac{5}{16}$ "	$\frac{9}{16}$ "	$\frac{5}{8}$ "	$\frac{15}{16}$ "
$\frac{3}{8}$ "	$\frac{11}{16}$ "	$\frac{3}{4}$ "	$1\frac{1}{8}$ "
$\frac{7}{16}$ "	$\frac{3}{4}$ "	$\frac{7}{8}$ "	$1\frac{1}{8}$ "
$\frac{1}{2}$ "	$\frac{13}{16}$ "	1 "	$1\frac{1}{4}$ "

The efficiency of the joint is the percentage of the strength of the solid plate that is retained in the joint, and it depends upon the kind of joint and method of construction.

If the thickness of the plate is more than $\frac{1}{2}$ in., the joint should always be of the double butt type.

The diameters of rivets, rivet holes, pitch and efficiency of joint, as given in the following Table 2, which

TABLE 2.

PROPORTIONS AND EFFICIENCIES OF RIVETED JOINTS.

	Inch.	Inch.	Inch.	Inch.	Inch.
Thickness of plate	$\frac{1}{4}$	$\frac{5}{16}$	$\frac{3}{8}$	$\frac{7}{16}$	$\frac{1}{2}$
Diameter of rivet.	$\frac{5}{8}$	$\frac{11}{16}$	$\frac{3}{4}$	$\frac{13}{16}$	$\frac{7}{8}$
Diameter of rivet-hole	$\frac{11}{16}$	$\frac{3}{4}$	$\frac{13}{16}$	$\frac{7}{8}$	$\frac{15}{16}$
Pitch for single riveting	2	$2\frac{1}{8}$	$2\frac{1}{8}$	$2\frac{3}{8}$	$2\frac{1}{4}$
Pitch for double riveting	3	$3\frac{1}{8}$	$3\frac{1}{4}$	$3\frac{3}{8}$	$3\frac{1}{2}$
Efficiency, single-riveted joint....	.66	.64	.62	.60	.58
Efficiency, double-riveted joint....	.77	.76	.75	.74	.73

was published in the "Locomotive" several years ago, were adopted at the time by some of the best establishments in the United States:

Concerning the proportions of double-riveted butt joints, Prof. Kent says: "Practically it may be said that we get a double-riveted butt joint of maximum strength by making the diameter of the rivet about 1.8 times the thickness of the plate, and making the pitch 4.1 times the diameter of the hole."

Table 3 as given below is condensed from the report of a test of double-riveted lap and butt joints. In this test the tensile strength of the plates was 56,000 to 58,000 pounds per square inch, and the shearing resistance of the rivets (steel) was about 50,000 pounds per square inch.

TABLE 3.

DIAMETER AND PITCH OF RIVETS—DOUBLE-RIVETED JOINT.

Kind of Joint.	Thickness of Plate.	Diameter of Rivet.	Ratio of Pitch to Diameter.
Lap	$\frac{3}{8}$ inch	0.8 inches	3.6 inches
Butt	$\frac{3}{8}$ "	0.7 "	3.9 "
Butt	$\frac{3}{4}$ "	1.1 "	4.0 "
Butt	1 "	1.3 "	3.9 "

Lloyd's rules, condensed, are as follows:

LLOYD'S RULES—THICKNESS OF PLATE AND DIAMETER OF RIVETS.

Thickness of Plate.	Diameter of Rivets.	Thickness of Plate.	Diameter of Rivets.
$\frac{3}{8}$ inch	$\frac{5}{8}$ inch	$\frac{3}{4}$ inch	$\frac{7}{8}$ inch
$\frac{7}{16}$ "	$\frac{5}{8}$ "	$\frac{13}{16}$ "	$\frac{7}{8}$ "
$\frac{1}{2}$ "	$\frac{3}{4}$ "	$\frac{7}{8}$ "	1 "
$\frac{9}{16}$ "	$\frac{3}{4}$ "	$\frac{15}{16}$ "	1 "
$\frac{5}{8}$ "	$\frac{3}{4}$ "	1 "	1 "
$\frac{11}{16}$ "	$\frac{7}{8}$ "		

The following Table 4 is condensed from one calculated by Prof. Kent, in which he assumes the shearing strength of the rivets to be four-fifths of the tensile strength of the plate per square inch, and the excess strength of the perforated plate to be 10 per cent.

TABLE 4.

Thickness of Plate.	Diameter of Hole.	—Pitch—		—Efficiency—	
		Single Riveting.	Double Riveting.	Single Riveting.	Double Riveting.
Inches	Inches	Inches	Inches	Per Cent	Per Cent
$\frac{3}{8}$	$\frac{7}{8}$	2.04	3.20	57.1	72.7
$\frac{7}{16}$	1	2.30	3.61	56.6	72.3
$\frac{1}{2}$	1	2.14	3.28	53.3	70.0
$\frac{1}{2}$	$1\frac{1}{8}$	2.57	4.01	56.2	72.0
$\frac{9}{16}$	1	2.01	3.03	50.4	67.0
$\frac{9}{16}$	$1\frac{1}{8}$	2.41	3.69	53.3	69.5
$\frac{9}{16}$	$1\frac{1}{4}$	2.83	4.42	55.9	71.5
$\frac{5}{8}$	1	1.91	2.82	47.7	64.6
$\frac{5}{8}$	$1\frac{1}{8}$	2.28	3.43	50.7	67.3
$\frac{5}{8}$	$1\frac{1}{4}$	2.67	4.10	53.3	69.5

Another table of joint efficiencies as given by Dr. Thurston is as follows, slightly condensed from the original calculation:

TABLE 5.

Single riveting

Plate thickness.	$\frac{5}{16}$ "	$\frac{3}{8}$ "	$\frac{7}{16}$ "	$\frac{1}{2}$ "	$\frac{5}{8}$ "	$\frac{3}{4}$ "	$\frac{7}{8}$ "	1"
Efficiency	.55	.55	.53	.52	.48	.47	.45	.43

Double riveting

Plate thickness.	$\frac{3}{8}$ "	$\frac{7}{16}$ "	$\frac{1}{2}$ "	$\frac{3}{4}$ "	$\frac{7}{8}$ "	1"
Efficiency	.73	.72	.71	.66	.61	.63

The author has been at considerable pains to compile Tables 6, 7 and 8, giving proportions and efficiencies of single lap, double lap and butt, and triple-riveted butt joints. The highest authorities have been consulted in the computation of these tables and great care exercised in the calculations.

TABLE 6.

PROPORTIONS OF SINGLE-RIVETED LAP JOINTS.

Thickness of Plate Inches.	Diameter of Rivet Inches.	Pitch of Rivet Inches.	Efficiency Per Cent.
5/16	9/16	1.13	50.5
"	5/8	1.33	53.3
"	11/16	1.55	55.7
3/8	3/4	1.60	53.3
"	7/8	2.04	57.1
7/16	7/8	1.87	53.2
"	1	2.30	56.6
1/2	1	2.14	53.3
"	1 1/8	2.57	56.2
9/16	1	2.01	50.4
"	1 1/8	2.41	53.3
"	1 1/4	2.83	55.9
5/8	1 1/8	2.28	50.7
"	1 1/4	2.67	53.3

It will be noticed that in single-riveted lap joints the highest efficiencies are attained when the diameter of the rivet hole is about $2\frac{1}{3}$ times the thickness of the plate, and the pitch of the rivet $2\frac{3}{8}$ times the diameter of the hole.

With the double-riveted joint it appears, according to Table 7, that in order to obtain the highest efficiency the joint should be designed so that the diameter of the

rivet hole will be from $1\frac{4}{5}$ to 2 times the thickness of plate, and the pitch should be from $3\frac{1}{3}$ to $3\frac{1}{2}$ times the diameter of the hole. Concerning the thickness of plates Dr. Thurston has this to say: "Very thin plates cannot be well caulked, and thick plates cannot be safely riveted. The limits are about $\frac{1}{4}$ of an inch for the lower limit, and $\frac{3}{4}$ of an inch for the higher limit." The riveting machine, however, overcomes the difficulty with very thick plates.

TABLE 7.

PROPORTIONS OF DOUBLE-RIVETED LAP AND BUTT JOINTS.

Thickness of Plate.	Diameter of Rivet.	Pitch of Rivet.	Efficiency.
$5/16$ inch	$9/16$ inch	1.71 inches	67.1 per cent
$5/16$ "	$5/8$ "	2.05 "	69.5 "
$3/8$ "	$3/4$ "	2.46 "	69.5 "
$3/8$ "	$7/8$ "	3.20 "	72.7 "
$7/16$ "	$3/4$ "	2.21 "	66.2 "
$7/16$ "	$7/8$ "	2.86 "	69.4 "
$7/16$ "	1 "	3.61 "	72.3 "
$1/2$ "	1 "	3.28 "	70.0 "
$1/2$ "	$1\frac{1}{8}$ "	4.01 "	72.0 "
$9/16$ "	1 "	3.03 "	67.0 "
$9/16$ "	$1\frac{1}{8}$ "	3.69 "	69.5 "
$9/16$ "	$1\frac{1}{4}$ "	4.42 "	71.5 "
$5/8$ "	$1\frac{1}{8}$ "	3.43 "	67.3 "
$5/8$ "	$1\frac{1}{4}$ "	4.10 "	69.5 "
$3/4$ "	1 "	2.50 "	72.0 "
$7/8$ "	$1\frac{1}{8}$ "	3.94 "	74.2 "
1 "	$1\frac{1}{4}$ "	4.10 "	76.1 "

The triple-riveted butt joint with two welts, one inside and one outside, has two rows of rivets in double shear, and one outer row in single shear on each side of the butt, the pitch of rivets in the outer rows being twice

the pitch of the inner rows. One of the welts is wide enough for the three rows of rivets each side of the butt, while the other welt takes in only the two close pitch rows.

When properly designed this form of joint has a high efficiency, and is to be relied upon. Table 8 gives proportions and efficiencies, and it will be noted that the highest degree of efficiency is shown when the diameter of rivet hole is from $1\frac{1}{4}$ to $1\frac{1}{2}$ times the thickness of plate, and the pitch of the rivets is from $3\frac{1}{2}$ to 4 times the diameter of the hole. This, of course, refers to the pitch of the close rows of rivets, and not the two outer rows.

TABLE 8.

PROPORTIONS OF TRIPLE-RIVETED BUTT JOINTS WITH INSIDE AND OUTSIDE WELT.

Thickness of Plate Inches,	Diameter of Rivet Inches.	Pitch of Rivet Inches.	Pitch of Outer Rows Inches.	Efficiency Per Cent.
$\frac{3}{8}$	13/16	3.25	6.5	84
7/16	13/16	3.25	6.5	85
$\frac{1}{2}$	13/16	3.25	6.5	83
9/16	$\frac{7}{8}$	3.50	7.0	84
$\frac{5}{8}$	1	3.50	7.0	86
$\frac{3}{4}$	1 1/16	3.50	7.0	85
$\frac{7}{8}$	$1\frac{1}{8}$	3.75	7.5	86
1	$1\frac{1}{4}$	3.87	7.7	84

The highest efficiency is attained in a riveted joint when the tensile strength of the rods from which the rivets are cut approaches that of the plates, and when the proportions of the joint are such that the tensile strength of the plates, the shearing strength of the rivets, and the crushing resistance of the rivets and plate, for a given section or unit strip, are as nearly equal as it is possible to secure them.

A few examples of calculations for efficiency will be given, taking the three forms of riveted joints in most common use. The following notation will be used throughout:

T. S.=Tensile strength of plate per square inch.

T=Thickness of plate.

C=Crushing resistance of plate and rivets.

A=Sectional area of rivets.

S=Shearing strength of rivets.

D=Diameter of hole (also diameter of rivets when driven).

P=Pitch of rivets.

In the calculations that follow T. S. will be assumed to be 60,000 lbs., S will be taken at 45,000 lbs., and the value of C may be assumed to be 90,000 to 95,000.

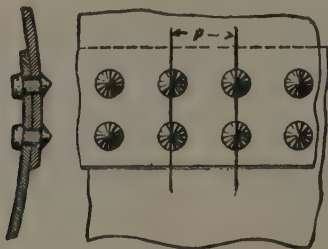


Fig. 25.

DOUBLE-RIVETED LAP AND BUTT JOINTS.

Fig. 25 shows a double-riveted lap joint. The style of riveting in this joint is what is known as chain riveting.

In case the rivets are staggered, the same rules for calculating the efficiency will hold as with chain rivet-

ing, for the reason that with either style of riveting the unit strip of plate has a width equal to the pitch or distance p . (Fig. 25.)

The dimensions of the joint under consideration are as follows: $P=3\frac{1}{4}$ in., $T=7/16$ in., $D=1$ in. (which is also diameter of driven rivet).

The strength of the unit strip of solid plate is $P \times T \times T.S.=85,312$.

The strength of net section of plate after drilling is $P-D \times T \times T.S.=59,062$.

The shearing resistance of two rivets is $2A \times S=70,686$.

The crushing resistance of rivets and plate is $D \times 2 \times T \times C=78,750$.

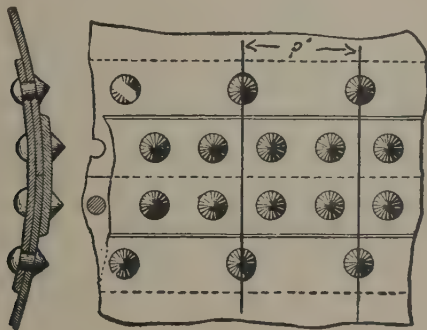


Fig. 26.

It thus appears that the weakest part of the joint is the net strip or section of plate, the strength of which is 59,062 and the efficiency $=59,062 \times 100 \div 85,312=69.2$ per cent.

A double-riveted butt joint is illustrated by Fig. 26, and the dimensions are as follows:

P, inner row of rivets= $2\frac{3}{4}$ in.

P', outer row of rivets= $5\frac{1}{2}$ in.

T of plate and butt straps= $7/16$ in.

D of hole and driven rivet=1 in.

Failure may occur in this joint in five distinct ways, which will be taken up in their order.

1. Tearing of the plate at the outer row of rivets. The net strength at this point is $P-D \times T \times T.S.$, which, expressed in plain figures, results as follows: $5.5-1 \times .4375 \times 60,000=118,125$.

2. Shearing two rivets in double shear and one in single shear. Should this occur, the two rivets in the inner row would be sheared on both sides of the plate, thus being in double shear. Opposed to this strain there are four sections of rivets, two for each rivet. Then at the outer row of rivets in the unit strip there is the area of one rivet in single shear to be added. The total resistance, therefore, is $5A \times S$ as follows: $.7854 \times 5 \times 45,000=176,715$.

3. The plate may tear at the inner row of rivets and shear one rivet in the outer row. The resistance in this case would be $P'-2D \times T \times T.S.+A \times S$ as follows: $5.5-2 \times .4375 \times 60,000+.7854 \times 45,000=127,218$.

4. Failure may occur by crushing in front of three rivets. Opposed to this is $3D \times T \times C$, or $1 \times 3 \times .4375 \times 95,000=124,687$.

5. Failure may occur by crushing in front of two rivets and shearing one. The resistance is represented by $2D \times T \times C+1A \times S$; expressed in figures, $1 \times 2 \times .4375 \times 95,000+.7854 \times 45,000=118,468$.

The strength of a solid strip of plate $5\frac{1}{2}$ in. wide before drilling is $P' \times T \times T.S.$, or $5.5 \times .4375 \times 60,000 = 144,375$, and the efficiency of the joint is $118,125 \times 100 \div 144,375 = 81.1$ per cent.

TRIPLE-RIVETED BUTT JOINT.

A triple-riveted butt joint is shown in Fig. 27, the dimensions of which are as follows:

$T = 7/16$ in., $D = 15/16$ in., $A = .69$ in., $P = 3\frac{3}{8}$ in., $P' = 6\frac{3}{4}$ in.

Failure may occur in this joint in either one of five ways.

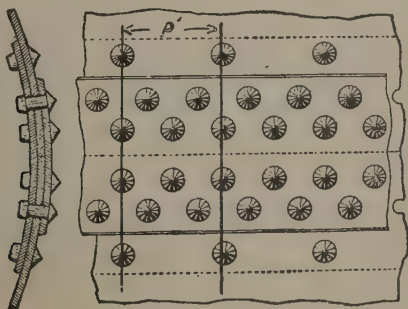


Fig. 27.

1. By tearing the plate at the outer row of rivets where the pitch is $6\frac{3}{4}$ in. The net strength of the unit strip at this point is $P' - D \times T \times T.S.$, found as follows: $6.75 - .9375 \times .4375 \times 60,000 = 152,578$.

2. By shearing four rivets in double shear and one in single shear. In this instance, of the four rivets in double shear, each one presents two sections, and the one in single shear presents one, thus making a total of

nine sections of rivets to be sheared, and the strength is $9A \times S$, or $.69 \times 9 \times 45,000 = 279,450$.

3. Rupture of the plate at the middle row of rivets and shearing one rivet. Opposed to this strain the strength is $P' - 2D \times T \times T.S. + 1A \times S$, equivalent to $6.75 - (.9375 \times 2) \times .4375 \times 60,000 + .69 \times 90,000 = 190,068$.

4. Crushing in front of four rivets and shearing one rivet. The resistance in this instance is $4D \times T \times C + 1A \times S$, or $.9375 \times 4 \times .4375 \times 90,000 + .69 \times 45,000 = 178,706$.

5. Failure may be caused by crushing in front of five rivets, four of which pass through both the inside and outside butt straps, while the fifth rivet passes through the inside strap only, and the resistance is $5D \times T \times C$, equivalent to $.9375 \times 5 \times 90,000 = 184,570$.

The strength of the unit strip of plate before drilling is $P' \times T \times T.S.$, or $6.75 \times .4375 \times 60,000 = 177,187$, and the efficiency is $152,578 \times 100 \div 177,187 = 86$ per cent.

With the constantly increasing demand for higher steam pressures, the necessity for higher efficiencies in the riveted joints of boilers becomes more apparent, and of late years quadruple and even quintuple-riveted butt joints have in many instances come into use. The quadruple butt joint when properly designed shows a high efficiency, in some cases as high as 94.6 per cent. Fig. 28 illustrates a joint of this kind, and the dimensions are as follows:

$$T = \frac{1}{2} \text{ in.}$$

$$D = \frac{15}{16} \text{ in.}$$

$$A = .69 \text{ in.}$$

$$P, \text{ inner rows} = 3\frac{3}{4} \text{ in.}$$

$$P', \text{ 1st outer row} = 7\frac{1}{2} \text{ in.}$$

$$P'', \text{ 2d outer row} = 15 \text{ in.}$$

The two inner rows of rivets extend through the main plate and both the inside and outside cover plates or butt straps.

The two outer rows reach through the main plate and inside cover plate only, the first outer row having twice the pitch of the inner rows, and the second outer row has twice the pitch of the first.

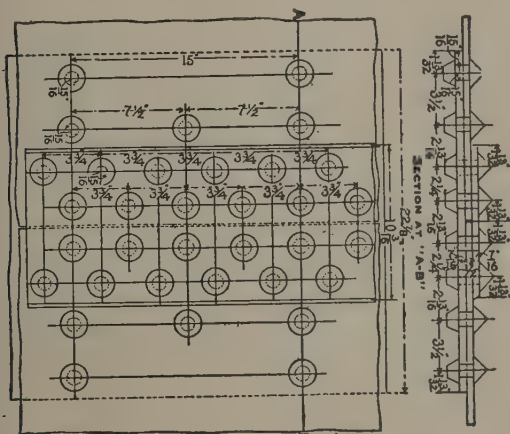


Fig. 28.

Taking a strip or section of plate 15 in. wide (pitch of outer row), there are four ways in which this joint may fail.

1. By tearing of the plate at the outer row of rivets. The resistance is $P'' - D \times T \times T.S.$, or $15 - .9375 \times .5 \times 60,000 = 421,875$.

2. By shearing eight rivets in double shear and three in single shear. The strength in resistance is $19A \times S$, or $.69 \times 19 \times 45,000 = 589,950$.

3. By tearing at inner rows of rivets and shearing three rivets. The resistance is $P'' - 4D \times T \times T.S. + 3A \times S$, or $15 - (.9375 \times 4) \times .5 \times 60,000 + .69 \times 3 \times 45,000 = 430,650$.

4. By tearing at the first outer row of rivets, where the pitch is $7\frac{1}{2}$ in., and shearing one rivet. The resistance is $P'' - 2D \times T \times T.S. + A \times S$, or $15 - (.9375 \times 2) \times .5 \times 60,000 + .69 \times 45,000 = 424,800$.

It appears that the weakest part of the joint is at the outer row of rivets, where the net strength is 421,875. The strength of the solid strip of plate 15 in. wide before drilling is $P'' \times T \times T.S.$, or $15 \times .5 \times 60,000 = 450,000$, and the efficiency is $421,875 \times 100 \div 450,000 = 93.7$ per cent.

Fig. 29 shows another style of quadruple-riveted butt joint. This joint is now used on nearly all high-grade boilers of the horizontal return-tubular type, and it marks about the practical limit of efficiency for riveted joints connecting plates of uniform thickness together. The methods of failure to be considered are practically the same as in the two preceding joints, except that there are more rivets concerned in the calculations:

- (1) Pulling apart of the sheets along net section A A.
- (2) Pulling apart of the sheet along section D E F G and shearing rivets A, B, C.
- (3) Pulling apart of sheet along section D E F G and crushing of rivets A, B, C in the strap.
- (4) Shearing rivets A, B, C in single shear and D, E, F, G, H, I, J, K in double shear.
- (5) Crushing of rivets D, E, F, G, H, I, J, K in plate and A, B, C in the strap.
- (6) Crushing of rivets D, E, F, G, H, I, J, K in the plate and shearing rivets A, B, C.

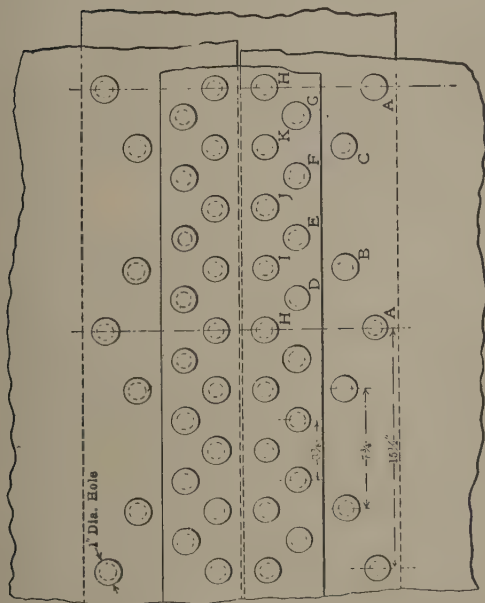
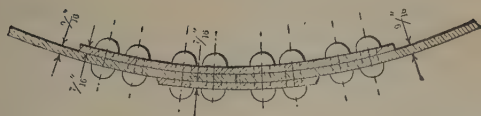


Fig. 29.

Using the numerical values previously given:

$$(1) (15.5-1) \times 0.5625 \times 55,000 = 448,580 \text{ pounds.}$$

$$(2) [(15.5-4) \times 0.5625 \times 55,000] + (3 \times 42,000 \times 0.7854) = 454,739 \text{ pounds.}$$

$$(3) [(15.5-4) \times 0.5625 \times 55,000] + (3 \times 0.4375 \times 1 \times 95,000) = 480,465 \text{ pounds.}$$

$$(4) (3 \times 42,000 \times 0.7854) + (8 \times 78,000 \times 0.7854) = 589,050 \text{ pounds.}$$

$$(5) (8 \times 0.5625 \times 1 \times 95,000) + (3 \times 0.4375 \times 1 \times 95,000) = 552,187 \text{ pounds.}$$

$$(6) (8 \times 0.5625 \times 1 \times 95,000) + (3 \times 42,000 \times 0.7854) = 526,461 \text{ pounds.}$$

The strength of the solid plate is

$$15.5 \times 0.5625 \times 55,000 = 479,528$$

pounds, and the failure of the sheet by pulling apart along the net section A A is the one that determines the efficiency of the joint, which is

$$\frac{448,580}{479,528} = 93.55 \text{ per cent.}$$

Staying Flat Surfaces. The proper staying or bracing of all flat surfaces in steam boilers is a highly important problem, and while there are various methods of bracing resorted to, still, as Dr. Peabody says, "the staying of a flat surface consists essentially in holding it against pressure at a series of isolated points which are arranged in regular or symmetrical pattern." The cylindrical shell of a boiler does not need bracing, for the very simple reason that the internal pressure tends to keep it cylindrical. On the contrary, the internal pressure has a constant tendency to bulge out the flat surface. Rule 2, Section 6, of the rules of the U. S. Supervising Inspectors provides as follows: "No braces or stays hereafter to be employed in the construction of boilers shall be allowed a greater strain than 6,000 lbs. per sq. in. of section."

The method to be employed in staying a boiler depends upon the type of boiler and the pressure to be

carried. Formerly when comparatively low pressures were used (60 to 75 lbs. per sq. in.) the diagonal crow foot brace was considered amply sufficient for staying the flat heads of boilers of the cylindrical tubular type, both above and below the tubes, but in the present age, when much higher pressures are demanded, through stay rods are largely employed. These are soft steel or iron rods $1\frac{1}{4}$ to 2 in. in diameter, extending through from head to head, with a pull at right angles to the plate, thus having a great advantage over the diagonal stay in that the pull on the diagonal stay per square inch of section is more than 5 per cent in excess of what a through stay would have to resist under the same conditions of pressure.

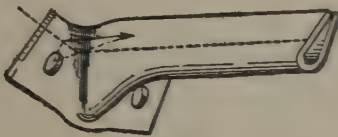


Fig. 30.

The weakest portion of the crow foot brace when in position is at the foot end, where it is connected to the head by two rivets. With a correctly designed brace the pull on these rivets is direct, and the tensile strength of the material needs to be considered only, but if the form of the brace is such as to bring the rivet holes above or below the center line of the brace, or if the rivets are pitched too far from the body of the brace, there will be a certain leverage exerted upon the rivets in addition to the direct pull. Fig. 30 shows a brace of incorrect design and Figs. 31 and 32 show braces designed along correct lines.

Other methods of staying, besides the crow foot brace and through stays, consist of gusset stays, and for locomotives, and other fire box boilers screwed stay bolts are employed to tie the fire box to the external shell. The holes for these stay bolts are punched or drilled before the fire box is put in place. After it is in and riveted along the lower edge to the foundation ring, or mud ring as it is sometimes called, a continuous thread is tapped in the holes in both the outside plate and the fire sheet by running a long tap through both plates. The



Fig. 31.



Fig. 32.

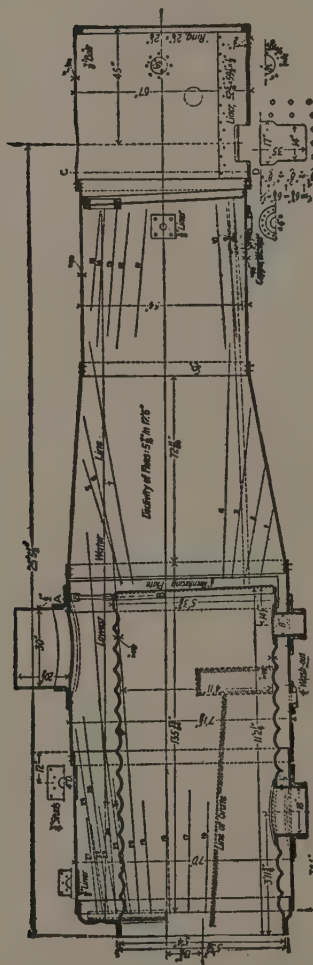
steel stay bolt is then screwed through the plates and allowed to project enough at each end to permit of its being riveted cold. Stay bolts are liable to be broken by the unequal expansion of the fire box and outer shell, and a small hole should be drilled in the center of the bolt, from the outer end nearly through to the inner end. Then in case a bolt breaks, steam or water will blow out through the small hole, and the break will be discovered at once. The problem of properly staying the flat crown

sheet of a horizontal fire box boiler, especially a locomotive boiler, is a very difficult one and has taxed the inventive genius of some of the most eminent engineers.

Before the invention of the Belpaire boiler, with its outside, or shell plate flat above the fire box, the only method of staying the crown sheet was by the use of cumbersome crown bars or double girders extending across the top of the crown sheet and supported at the ends by special castings that rest on the edges of the side sheets and on the flange of the crown sheet. At intervals of 4 or 5 in. crown bolts are placed, having the head inside the fire box and the nut bearing on a plate on top of the girder. There is also a thimble for each bolt to pass through, between the top of the crown sheet and the girder. These thimbles maintain the proper distance between the crown sheet and girder and allow the water to circulate freely.

The Belpaire fire box dispenses with girders, and permits the use of through stays from the top of the flat outside plate through the crown sheet and secured at each end by nuts and copper washers.

For simplicity of construction and great strength the cylindrical form of fire box known as the Morison corrugated furnace has proved to be very successful. This form of fire box was in 1899 applied to a locomotive by Mr. Cornelius Vanderbilt, at the time assistant superintendent of motive power of the New York Central & Hudson River R. R. This furnace was rolled of $\frac{3}{4}$ -in. steel, is 59 in. internal diameter and 11 ft. $2\frac{1}{4}$ in. in length. It was tested under an external pressure of 500 lbs. per sq. in. before being placed in the boiler. It is carried at the front end by a row of radial sling stays from the outside plate, and supported at the rear by the



Fl. 33.

back head. Figs 33 and 34 show respectively a sectional view, and an end elevation of this boiler. It will be seen at once that the question of stays for a fire box of this type becomes very simple. The boiler has proved to be so satisfactory that the company has since had five more of the same type constructed.

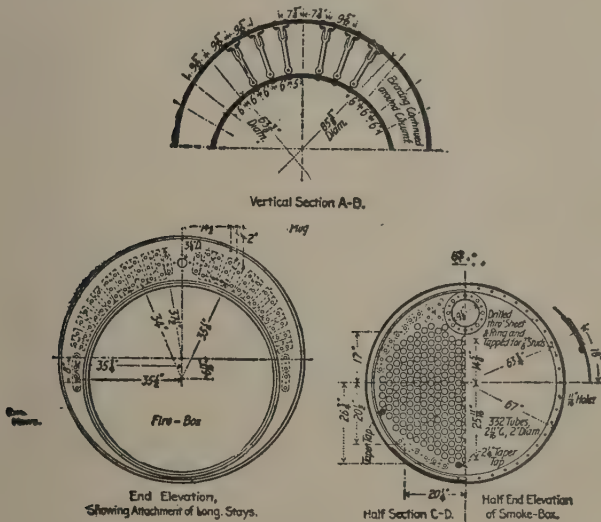


Fig. 34.

Gusset stays are used mainly in boilers of the Lancashire model, and are triangular-shaped plates sheared to the proper form and having two angle irons riveted to the edges that come against the shell and the head. The angle irons are then riveted to the shell and the flat head. This form of brace is simple and solid, but its chief defect is, that it is very rigid and does not allow

for the unequal expansion of the internal furnace flues and the shell. Fig. 35 illustrates a gusset stay and the method of applying it.

Coming now to through stay rods, it is safe to say that whenever, and wherever it is possible to apply them they should be used. In all cases they should be placed far enough apart to allow a man to pass between them for the purposes of inspection and washing out of the boiler. Through stay rods are usually spaced 14 in.

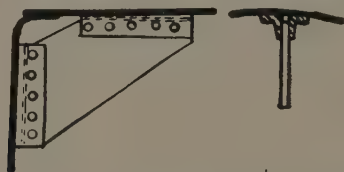


Fig. 35.

apart horizontally, and about the same distance vertically. The ends, as far back as the threads run, are swaged larger than the body, so that the diameter at the bottom of the thread is greater than the diameter of the body. There are several methods of applying through stays. One of the most common, especially for land boilers, is to allow the ends of the rod to project through the plates to be stayed, and holding them in place by a nut and copper washer, both inside and outside the plate. Another and still better plan is to rivet 6-in. channel bars across the head, inside above the tubes, the number of bars depending upon the height of the segment to be stayed.

The channel bars are drilled to correspond with the holes that are drilled in the plate to receive the stay rods, which latter are then secured by inside and outside nuts and copper washers. These channel bars act as girders,

and serve to greatly strengthen the head or flat plate. Fig. 36 will serve to illustrate this method.

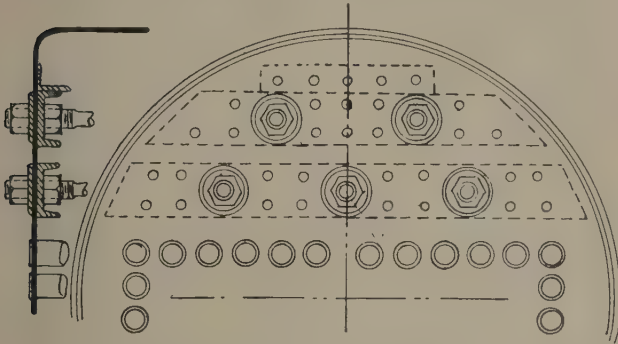


Fig. 36.

Sometimes a combination of channel bar and diagonal crow foot braces is used, as shown by Fig. 37.

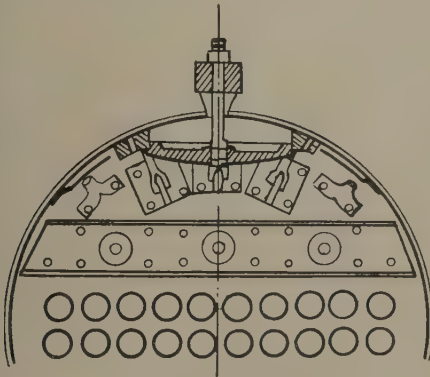


Fig. 37.

A good form of diagonal crow foot stay is obtained by using double crow feet, made of pieces of boiler plate

bent as shown by Fig. 38 and riveted to the plate by four rivets. A hole is drilled through the body of the crow foot, and a bolt passing through this secures the forked end of the stay.

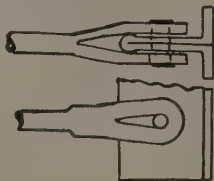


Fig. 38.

Another method of securing through stays to the heads is shown by Fig. 39 and is applied where too many stay

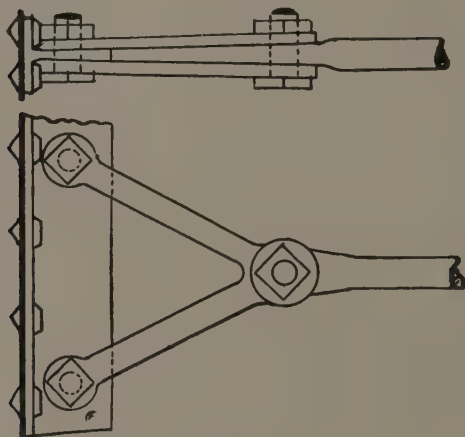


Fig. 39.

rods would be required to connect all the points to be stayed. A tee iron is first riveted to the flat plate to be stayed, and two V-shaped forgings are bolted to it as

shown. The through stay is then bolted to the forgings, and thus two points in the flat head are supported by one stay. It will readily be seen that this method will reduce the number of through stay rods required.

Calculating the Strength of Stayed Surfaces. In calculations for ascertaining the strength of stayed surfaces, or for finding the number of stays required for any given flat surface in a boiler, the working pressure being known, it must be remembered that each stay is subjected to the pressure on an area bounded by lines drawn midway between it and its neighbors. Therefore the area in square inches, of the surface to be supported by each stay, equals the square of the pitch or distance in inches between centers of the points of connection of the stays to the flat plate. Thus, suppose the stays in a certain boiler are spaced 8 in. apart, the area sustained by each stay $= 8 \times 8 = 64$ sq. in., or assume the stay bolts in a locomotive fire box to be pitched $4\frac{1}{2}$ in. each way, the area supported by each stay bolt $= 4\frac{1}{2} \times 4\frac{1}{2} = 20\frac{1}{4}$ sq. in. Again taking through stay rods, suppose, for example, the through stays shown in Fig 36 to be spaced 15 in. horizontally and 14 in. vertically, the area supported by each stay $= 15 \times 14 = 210$ sq. in.

The minimum factor of safety for stays, stay bolts and braces is 8, and this factor should enter into all computations of the strength of stayed surfaces.

The pitch for stays depends upon the thickness of the plate to be supported, and the maximum pressure to be carried.

In computing the total area of the stayed surface it is safe to assume that the flange of the plate, where it is riveted to the shell, sufficiently strengthens the plate

for a distance of 2 in. from the shell, also that the tubes act as stays for a space of 2 in. above the top row. Therefore the area of that portion of the flat head or plate bounded by an imaginary line drawn at a distance of 2 in. from the shell and the same distance from the last row of tubes is the area to be stayed. This surface may be in the form of a segment of a circle, as with a horizontal cylindrical boiler, or it may be rectangular in shape, as in the case of a locomotive, or other fire box boiler. Other forms of stayed surfaces are often encountered, but in general the rules applicable to segments or rectangular figures will suffice for ascertaining the areas.

The method of finding the area of the segmental portion of the head above the tubes is as follows, using Table 9. The diameter of the circle and the rise or height of the segment being known, the area of the segment may be found by the following rule:

Rule. Divide the height of the segment by the diameter of the circle. Then find the decimal opposite this ratio in the column headed "Area." Multiply this area by the square of the diameter. The result is the required area.

Example. Diameter of circle=72 in. Height of segment=25 in. $25 \div 72 = .347$, which will be found in the column headed "Ratio," and the area opposite this is .24212. Then $.24212 \times 72 \times 72 = 1,255$ sq. in., area of segment.

TABLE 9.
AREAS OF SEGMENTS OF A CIRCLE.

Ratio	Area	Ratio	Area	Ratio	Area	Ratio	Area
.2	.11182	.243	.14751	.286	.18542	.329	.22509
.201	.11262	.244	.14837	.287	.18633	.33	.22603
.202	.11343	.245	.14923	.288	.18723	.331	.22697
.203	.11423	.246	.15009	.289	.18814	.332	.22792
.204	.11504	.247	.15095	.29	.18905	.333	.22886
.205	.11584	.248	.15182	.291	.18996	.334	.22980
.206	.11665	.249	.15268	.292	.19086	.335	.23074
.207	.11746	.25	.15355	.293	.19177	.336	.23169
.208	.11827	.251	.15441	.294	.19268	.337	.23263
.209	.11908	.252	.15528	.295	.19360	.338	.23358
.21	.11990	.253	.15615	.296	.19451	.339	.23453
.211	.12071	.254	.15702	.297	.19542	.34	.23547
.212	.12153	.255	.15789	.298	.19634	.341	.23642
.213	.12235	.256	.15876	.299	.19725	.342	.23737
.214	.12317	.257	.15964	.3	.19817	.343	.23832
.215	.12399	.258	.16051	.301	.19908	.344	.23927
.216	.12481	.259	.16139	.302	.20000	.345	.24022
.217	.12563	.26	.16226	.303	.20092	.346	.24117
.218	.12646	.261	.16314	.304	.20184	.347	.24212
.219	.12729	.262	.16402	.305	.20276	.348	.24307
.22	.12811	.263	.16490	.306	.20368	.349	.24403
.221	.12894	.264	.16578	.307	.20460	.35	.24498
.222	.12977	.265	.16666	.308	.20553	.351	.24593
.223	.13060	.266	.16755	.309	.20645	.352	.24689
.224	.13144	.267	.16843	.31	.20738	.353	.24784
.225	.13227	.268	.16932	.311	.20830	.354	.24880
.226	.13311	.269	.17020	.312	.20923	.355	.24976
.227	.13395	.27	.17109	.313	.21015	.356	.25071
.228	.13478	.271	.17198	.314	.21108	.357	.25167
.229	.13562	.272	.17287	.315	.21201	.358	.25263
.23	.13646	.273	.17376	.316	.21294	.359	.25359
.231	.13731	.274	.17465	.317	.21387	.36	.25455
.232	.13815	.275	.17554	.318	.21480	.361	.25551
.233	.13900	.276	.17644	.319	.21573	.362	.25647
.234	.13984	.277	.17733	.32	.21667	.363	.25743
.235	.14069	.278	.17823	.321	.21760	.364	.25839
.236	.14154	.279	.17912	.322	.21853	.365	.25936
.237	.14239	.280	.18002	.323	.21947	.366	.26032
.238	.14324	.281	.18092	.324	.22040	.367	.26128
.239	.14409	.282	.18182	.325	.22134	.368	.26225
.24	.14494	.283	.18272	.326	.22228	.369	.26321
.241	.14580	.284	.18362	.327	.22322	.37	.26418
.242	.14666	.285	.18452	.328	.22415	.371	.26514

TABLE 9—CONTINUED.

Ratio	Area	Ratio	Area	Ratio	Area	Ratio	Area
.372	.26611	.405	.29827	.438	.33086	.471	.36373
.373	.26708	.406	.29926	.439	.33185	.472	.36471
.374	.26805	.407	.30024	.44	.33284	.473	.36571
.375	.26901	.408	.30122	.441	.33384	.474	.36671
.376	.26998	.409	.30220	.442	.33483	.475	.36771
.377	.27095	.41	.30319	.443	.33582	.476	.36871
.378	.27192	.411	.30417	.444	.33682	.477	.36971
.379	.27289	.412	.30516	.445	.33781	.478	.37071
.38	.27386	.413	.30614	.446	.33880	.479	.37171
.381	.27483	.414	.30712	.447	.33980	.48	.37270
.382	.27580	.415	.30811	.448	.34079	.481	.37370
.383	.27678	.416	.30910	.449	.34179	.482	.37470
.384	.27775	.417	.31008	.45	.34278	.483	.37570
.385	.27872	.418	.31107	.451	.34378	.484	.37670
.386	.27969	.419	.31205	.452	.34477	.485	.37770
.387	.28067	.42	.31304	.453	.34577	.486	.37870
.388	.28164	.421	.31403	.454	.34676	.487	.37970
.389	.28262	.422	.31502	.455	.34776	.488	.38070
.39	.28359	.423	.31600	.456	.34876	.489	.38170
.391	.28457	.424	.31699	.457	.34975	.49	.38270
.392	.28554	.425	.31798	.458	.35075	.491	.38370
.393	.28652	.426	.31897	.459	.35175	.492	.38470
.394	.28750	.427	.31996	.46	.35274	.493	.38570
.395	.28848	.428	.32095	.461	.35374	.494	.38670
.396	.28945	.429	.32194	.462	.35474	.495	.38770
.397	.29043	.43	.32293	.463	.35573	.496	.38870
.398	.29141	.431	.32392	.464	.35673	.497	.38970
.399	.29239	.432	.32491	.465	.35773	.498	.39070
.4	.29337	.433	.32590	.466	.35873	.499	.39170
.401	.29435	.434	.32689	.467	.35972	.5	.39270
.402	.29533	.435	.32788	.468	.36072		
.403	.29631	.436	.32887	.469	.36172		
.404	.29729	.437	.32987	.47	.36272		

The following examples of calculating the number of braces, and the spacing of the same will serve to make the matter plain.

A boiler is 66 in. in diameter, the working pressure is 100 lbs. per sq. in. The distance from the top row of tubes to the shell is 25 in. Required, the number of

diagonal crow foot braces that will be needed to support the heads above the tubes, also the sectional area of each brace. The thickness of the heads is $\frac{5}{8}$ in. and the T.S.=55,000 lbs. per sq. in.

Assume the head to be sufficiently strengthened by the flange for a distance of 2 in. from the shell, the diameter of the circle of which the segment above the tubes requires to be stayed is reduced by $2+2=4$ in. and will therefore be $66-4=62$ in. The rise or height of the segment above the tubes is $25-4=21$ in. Required, the area. $21 \div 62=.338$. Looking down the column headed "Ratio" in Table 9, area opposite .338 is .23358. Area of segment= $.23358 \times 62 \times 62=897.88$ sq. in. The total pressure on this area will be $897.88 \times 100=89,788$ lbs.

Assume the braces to be made of $1\frac{1}{8}$ in. round steel having a T.S. of 50,000 lbs. per sq. in. and to be designed in such a manner as to allow for loss of material in drilling the rivet holes in the crow feet. Each brace will have a sectional area of .994 sq. in., and using 8 as a factor of safety, the strength or safe holding power of each stay may be found as follows: $.994 \times 50,000 \div 8=6,212$ lbs., and the number of stays required= $89,788$ lbs. (total pressure) divided by 6,212 lbs. (strength of each stay)=14.5, or in round numbers 15. If the stays are made of flat bars of steel the sectional area should equal that of the round stays, and the dimensions of the crow feet of all stays should be such as to retain the full sectional area of the body after the rivet holes are drilled.

Each stay is connected to the plate by two $\frac{7}{8}$ -in. rivets, having a T.S. of 55,000 lbs. per sq. in. and a shearing strength of 45,000 lbs. per sq. in. These rivets are capable of resisting a direct pull of 10,818 lbs., using 5

as a factor of safety; ascertained as follows: $2A \times 45,000 \div 5 = 10,818 =$ strength of two rivets. They are also subjected to a crushing strain, and the resistance to this is $D \times C \div 5$, which expressed in figures is $.875 \times 90,000 \div 5 = 15,750$ lbs.

The proper spacing comes next, and is arrived at in the following manner:

Area to be stayed $= 897.88$ sq. in.

Number of stays $= 15$.

Area supported by each stay $= 897.88 \div 15 = 59.8$ sq. in.

The square root of $59.8 = 7.75$ nearly, which is the distance in inches each way that the stays should be spaced, center to center.

If through stay rods are used in place of diagonal braces for staying the boiler under consideration, the number and diameter of the rods may be ascertained by the following method:

Assuming the heads to be supported by channel bars, as previously described, and that the stays are pitched 14 in. apart horizontally and 13 in. vertically, each stay would be required to support an area of $14 \times 13 = 182$ sq. in., and the number of stays would be $897.88 \div 182 = 4.9$, in round numbers 5. See Fig. 36. The pressure being 100 lbs. per sq. in., the total stress on each stay $= 182 \times 100 = 18,200$ lbs. Assume the stay rods to be of soft steel having a T.S. of 50,000 lbs. per sq. in., and using a factor of safety of 8, the sectional area required for each stay will be found as follows: $18,200 \times 8 \div 50,000 = 2.9$ sq. in., and the diameter will be found as follows: $2.9 \div .7854 = 3.69$, which is the square of the diameter, and the square root of $3.69 = 1.9$ in., or practically 2 in. The same methods of calculation are ap-

plicable to the staying of the heads below the tubes, also for stay bolts in fire box boilers.

Strength of Unstayed Surfaces. A simple rule for finding the bursting pressure of unstayed flat surfaces is that of Mr. Nichols, published in the "Locomotive," February, 1890, and quoted by Prof. Kent in his "Pocket-book." The rule is as follows: "Multiply the thickness of the plate in inches by ten times the tensile strength of the material used, and divide the product by the area of the head in square inches." Thus,

Diameter of head = 66 in.

Thickness of head = $\frac{5}{8}$ in.

Tensile strength = 55,000 lbs.

Area of head = 3,421 sq. in.

$\frac{5}{8} \times 55,000 \times 10 \div 3,421 = 100$, which is the number of pounds pressure per square inch under which the unstayed head would bulge.

If we use a factor of safety of 8, the safe working pressure would be $100 \div 8 = 12.5$ lbs. per sq. in., but as the strength of the unstayed head is at best an uncertain quantity it has not been considered in the foregoing calculations for bracing, except as regards that portion of it that is strengthened by the flange.

In all calculations for the strength of stayed surfaces, and especially where diagonal crow foot stays are used, the strength of the rivets connecting the stay to the flat plate must be carefully considered. A large factor of safety, never less than 8, should be used, and the cross section of that portion of the foot of the stay through which the rivet holes are drilled should be large enough, after deducting the diameter of the hole, to equal the sectional area of the body of the stay.

Dished Heads. In boiler work where it is possible to use dished, or "bumped up" heads as they are sometimes called, this type of head is rapidly coming into use. Dished heads may be used in the construction of steam drums, also in many cases for dome-covers, thus obviating the necessity of bracing. The maximum depth of dish, as adopted by steel plate manufacturers April 4, 1901, is $\frac{1}{8}$ of the diameter of the head when flanged, and if the tensile strength and quality of the plate from which the heads are made are the same as those of the shell plate, the dished head becomes as strong as the shell, provided the head has the same thickness, or is slightly thicker than the shell plate.

Welded Seams. A few boiler manufacturers have succeeded in making welded seams, thus dispensing with the time-honored custom of riveting the plates together. A good welded joint approaches more nearly to the full strength of the material than can possibly be attained by rivets, no matter how correctly designed the riveted joint may be. The weld also dispenses with the necessity of caulking, and a boiler having a perfectly smooth surface inside, such as would be afforded by welded seams, would certainly be much less liable to collect scale and sediment than would one with riveted joints. But in order to make a success of welded seams the material used must be of the best possible quality, and great care and skill are required in the work.

The Continental Iron Works of Brooklyn, New York, exhibited at the St. Louis World's Fair in 1904, a welded steel plate soda pulp digester without a single riveted joint. The dimensions of this vessel, which may be likened to a cylinder boiler without flues, were as follows: Thickness of plate, $\frac{3}{4}$ in.; diameter, 9 ft.;

length, 43 ft. The heads were dished to the standard depth. The safe working pressure was 125 lbs. per sq. in. It appears not only possible, but probable, that the process of welding boiler joints may in time supplant the older custom of riveting.

QUESTIONS.

46. What principles should govern the design and construction of steam boilers?

Ans. First: They should be absolutely safe. Second: They should be economical in the consumption of fuel. Third: They should be capable of furnishing dry steam.

47. What is meant by the term tensile strength as applied to boiler material?

Ans. The number of pounds of pull that would be required to break a bar of the material in the direction of its length.

48. What is liable to occur in case the tensile strength is too high?

Ans. Cracking of the sheets, also certain changes in the physical properties of the metal.

49. Which are the stronger, punched or drilled plates?

Ans. If the material is good soft steel, punched plates show a greater shearing resistance.

50. What should be the tensile strength of rivet iron?

Ans. About 60,000 lbs. per sq. in.

51. What is a good test for a $\frac{5}{8}$ -in. rivet?

Ans. It should stand being doubled up and hammered together without being fractured.

52. What is the shearing resistance of iron rivets?

Ans. About 85 per cent of the original bar.

53. What is the shearing resistance of steel rivets?

Ans. 77 per cent of the original bar.

54. What is meant by efficiency of the joint?

Ans. The percentage of strength of the solid plate, that is retained in the joint.

55. What should be the style of joint with sheets thicker than $\frac{1}{2}$ in.?

Ans. It should be a double butt joint.

56. What should be the ratio of diameter of rivet to thickness of plate for double butt joints?

Ans. The diameter of the rivet should be about 1.8 times the thickness of sheet?

57. What should be the pitch of rivets?

Ans. Three and one-half to four times the diameter of the hole.

58. Describe the triple riveted butt joint.

Ans. It has two welts or straps, one inside, and one outside.

59. Is this a good form of joint?

Ans. It is.

60. What type of joint gives the highest efficiency?

Ans. A joint in which the tensile strength of the rods from which the rivets are cut approaches that of the plates, and when the proportions of the joint are such, that the tensile strength of the rivets, and the crushing resistance of the rivets and plate, for a given, or unit strip, are as nearly equal as it is possible to make them.

61. In how many ways may failure occur in a double riveted butt joint?

Ans. In five distinct ways.

62. Name the first manner of failure.

Ans. Tearing of the plate at outer row of rivets.

63. What is the second?

Ans. Shearing two rivets in double shear, and one in single shear.

64. What is the third manner of failure?

Ans. Tearing of the plate at inner row of rivets, and shearing one rivet in the outer row.

65. Describe the fourth method of failure?

Ans. Crushing in front of three rivets.

66. What is the fifth manner of failure?

Ans. Crushing in front of two rivets, and shearing one.

67. How may a triple riveted butt joint fail?

Ans. First: By tearing the plate at the outer row of rivets. Second: By shearing four rivets in double shear, and one in single shear. Third: Rupture of the plate at the middle row of rivets, and shearing one rivet. Fourth: Crushing in front of four rivets, and shearing one rivet.

68. What is the efficiency of the quadruple riveted butt joint?

Ans. In some cases as high as 94 per cent.

69. In what four ways may failure occur in this type of joint?

Ans. First: By tearing the plate at the outer row of rivets. Second: By shearing eight rivets in double shear, and three in single shear. Third: By tearing at inner row of rivets, and shearing three rivets. Fourth: By tearing at first outer row of rivets where the pitch is $7\frac{1}{2}$ in.

70. What is implied in the staying of a flat surface?

Ans. Holding it against pressure at a series of isolated points, which are arranged in symmetrical order.

71. Does the cylindrical shell of a boiler need bracing?

Ans. No.

72. Why is this?

Ans. Because the internal pressure tends to keep it cylindrical.

73. How are the heads sometimes stayed?

Ans. By through stay rods of soft steel, or iron $1\frac{1}{4}$ or 2 in. in diameter extending through from head to head.

74. What advantage has this form of stays?

Ans. The pull is at right angles to the plate.

75. What other methods of bracing the heads of high pressure boilers are used?

Ans. Gusset stays, and dished heads.

76. What is the minimum factor of safety for stays, and braces?

Ans. Eight.

77. Give a simple rule for finding the bursting pressure of unstayed flat surfaces?

Ans. Multiply the thickness of the plate in inches by ten times the tensile strength and divide the product by the area of the surface in square inches.

CARE AND OPERATION OF BOILERS.

Duties. The first act of the engineer on entering his boiler-room when he goes on duty should be to ascertain the exact height of the water in his boilers. This he can do by opening the valve in the drain pipe of the water column, allowing it to blow out freely for a few seconds, then close it tight and allow the water to settle back in the glass. This should be done with each boiler under steam, not only once, but several times during the day. No engineer should be satisfied with a general squint along the line of gauge glasses, but he should either go himself, or else instruct his fireman, or water tender to make the rounds of each boiler and be sure that the water is all right.

The instructions regarding the cleaning of fires, and firing, refer particularly to hand fired boilers. Mechanical stokers will be taken up in their regular order.

The next thing to be looked after is the fire. If the plant is run continuously day and night it is the duty of the firemen coming off watch to have the fires clean, the ash pits all cleaned out, a good supply of coal on the floor, and everything in good order for the oncoming force. A good fireman will take pride in always leaving things in neat shape for the man who is to relieve him.

Cleaning Fires. With some varieties of coal this is a comparatively easy task, especially if the boilers are fitted with shaking grates. With a coal that does not form a clinker on the grate bars, the fires can be kept

in good condition by cleaning them twice or three times in twenty-four hours, as the larger part of the loose ashes and noncombustible can be gotten rid of by shaking the grates and using the slice bar at intervals more

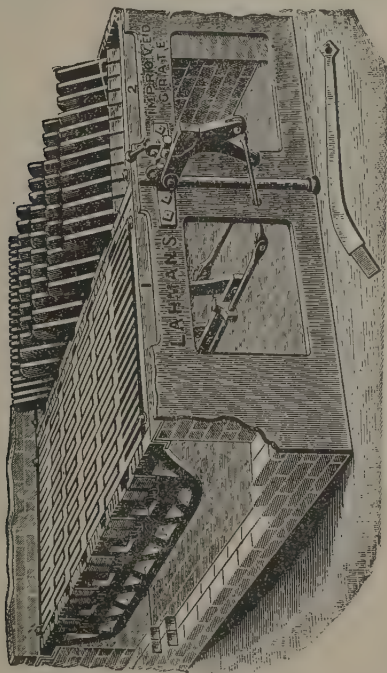


Fig. 40. Lehman's Grate.

or less frequent; but such coals are generally considered too expensive to use in the ordinary manufacturing plant, and cheaper grades are substituted.

Fire Tools. For cleaning fires successfully and quickly, the following tools should be provided; a slice

bar, a fire hook, a heavy iron or steel hoe, and a light hoe for cleaning the ash-pit. It is unnecessary to describe these tools, as they are familiar to all engineers. A suggestion as to the kind of handles with which they should be fitted may be of benefit. The working ends of the aforesaid tools having been made and each welded to a bar of 1 or $1\frac{1}{8}$ in. round iron and 10 or 12 in. long, take pieces of 1 or $1\frac{1}{4}$ in. iron pipe cut to the length desired for the handles and weld the shanks of the tools to them. To the other end of the pipe weld a handle made of round iron somewhat smaller than the shank. By using pipe handles the weight of the tools is considerably lessened, and they will still be sufficiently strong. The labor of cleaning the fire will thus be greatly lightened. When a fire shows signs of being foul and choked with clinker, preparations should be made at once for cleaning it by allowing one side to burn down as low as possible, putting fresh coal on the other side alone. When the first side has burned as low as it can without danger of letting the steam pressure fall too much, take the slice bar and run it in along the side of the furnace on top of the clinker and back to near the bridge wall, then using the door jamb as a fulcrum, give it a quick strong sweep across the fire and the greater part of the live coals will be pushed over to the other side. What remains of the coal not yet consumed can be pulled out upon the floor with the light hoe and shoveled to one side, to be thrown back into the furnace after the clinker is taken out. Having now disposed of the live coal, take the slice bar and run it along on top of the grates, loosening and breaking up the clinker thoroughly, after which take the heavy hoe and pull it all out on the floor. A helper

should be ready with a pail of water, or, what is still better, a small rubber hose connected to a cold water pipe running along the boiler fronts for this purpose, and put on just enough water to quench the intense heat of the red hot clinker as it lies on the floor. When the grates are cleaned, close the door, and with the slice

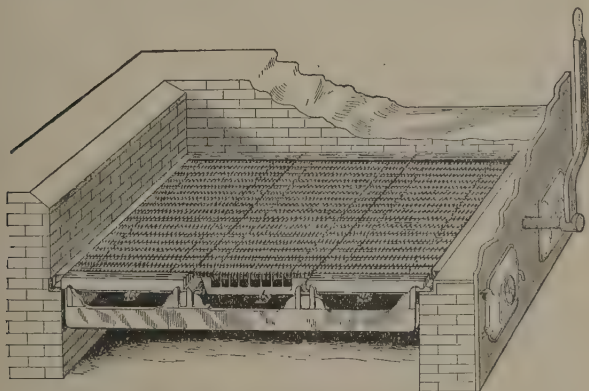


Fig. 41. Martin Anti-Friction Rocking Grates.

bar in the other side push all the live coal over to the side just cleaned, where it should be leveled off and fresh coal added. After this has become ignited, treat the other side in the same way. An expert fireman will thus clean a fire with very little loss in steam pressure, and practically no waste of coal.

DISPOSAL OF THE ASHES.

The problem of disposing of the ashes in large power plants is quite a serious one, and any device that tends to lessen the cost of labor, and shorten the time con-

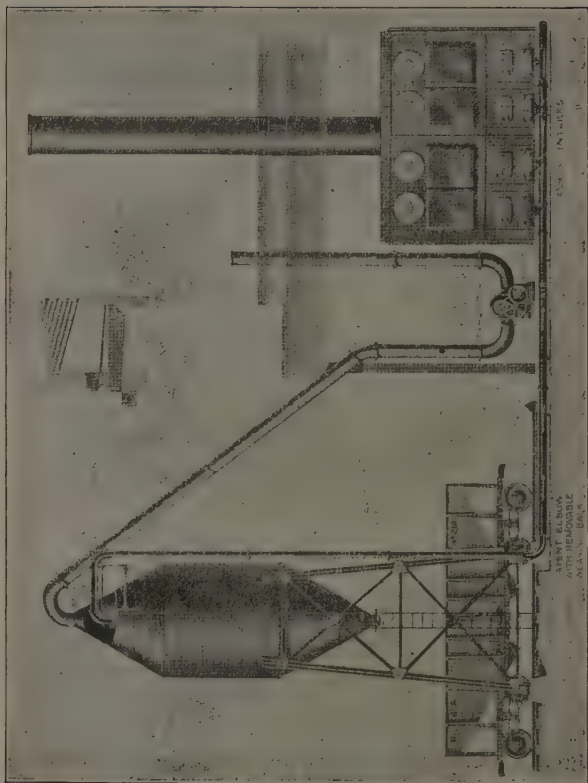


Fig. 42. Diagram Showing a Complete Suction Conveyor System as Applied to Handling Ashes From Boilers.

sumed in conveying the ashes from the boiler room certainly merits the attention of chief engineers.

The suction conveyor system of the Darley Engineering Company, Chicago, a general view of which is shown in Fig. 42, consists essentially of four parts, as follows:

- | | |
|-------------------|---------------|
| 1. Conveyor Pipe. | 3. Exhauster. |
| 2. Separator. | 4. Water Jet. |

The conveyor pipe line is made in three sizes and capacities, namely, 6", 8" and 10". of iron or steel pipe. The conveyor pipe, as far as possible, is run in straight lines. It is generally placed beneath the surface, but can be elevated or run anywhere to suit conditions.

The separator, which is in reality an expansion chamber, also serves as a storage tank for storing the conveyed material. This separator is always placed at the end of the conveyor run. It serves to catch the material conveyed and hold it until it can be drawn (by gravity through an under-cut gate) into cars, carts or barges. This separator is located in the most convenient position for the purpose. The separator can be made any size, to hold any predetermined amount of material, or for a time run of the conveyor of any fixed duration. These separators can also be mounted over bins or bunkers of large or small size, if such additional bin storage capacity is required. Separators are generally built of cylindrical form and of steel plate construction with a cone top and bottom. For certain work and on small sized plants, they can be made rectangular or of an irregular shape. They can also be made of concrete, either square or round, with small cone top of steel plate. They are always made water tight.

The Exhauster (Figs. 43 and 44) consists essentially

of a rotating impeller, surrounded by a suitable case, with an intake air opening at the center, and a discharge opening at the circumference. In appearance, it is similar to the centrifugal pump. The efficiency depends largely upon the design of the impeller and casing, and on the proper shaping of these parts. In actual

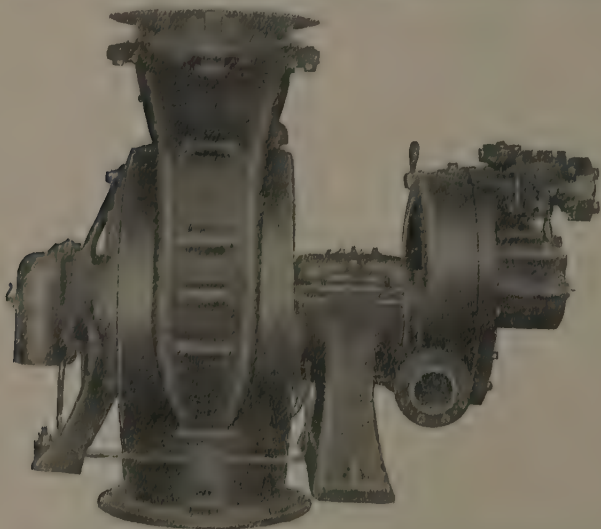


Fig. 43. Ash Conveyor-Exhauster Set driven by Steam Turbine.

practice some conditions call for other types. For instance, for a small, simple layout, an ordinary exhaust fan can be used, whereas for certain complicated and extensive work, a cycloidal blower is best adapted for the purpose. It can be either steam, or motor driven for alternating or direct current. These machines are very strong in construction, and will oper-

are under adverse conditions with a very low cost for maintenance, and are especially suitable for this class of work.

Just before entering the separator, the conveyed material passes through a water jet located in the conveyor pipe. This jet is composed of $\frac{1}{8}$ " holes, spaced 1" centers, and serves two purposes. *i. e.*, it takes the heat out of the hot material, such as ashes, etc., and eliminates all dust when the material is dusty. In this way all dust is kept out of the exhaustor.

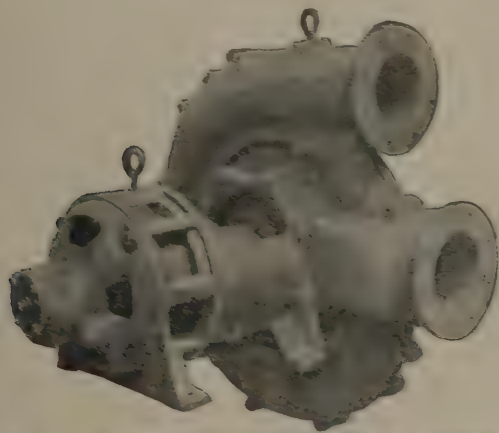


Fig. 44. Ash Conveyor—Exhaustor Set driven by Induction Motor

In the case of an ashes conveyor, the intakes are placed in front of the ash pits of the boilers, or at any other desired place, and the ashes are loaded or shoveled from the ash pits into these intakes, whence they disappear through the pipe line at a higher velocity. The

conveyor pipe line will take them away as fast as they are fed to the intakes. When not in use, a cast iron cover is placed over these intakes. These covers are lifted off when material is to be fed to the conveyor.

The size of the intake opening is slightly smaller in diameter than that of the pipe line, so that any piece of material that passes the intake opening will be conveyed freely through the conveyor pipe.

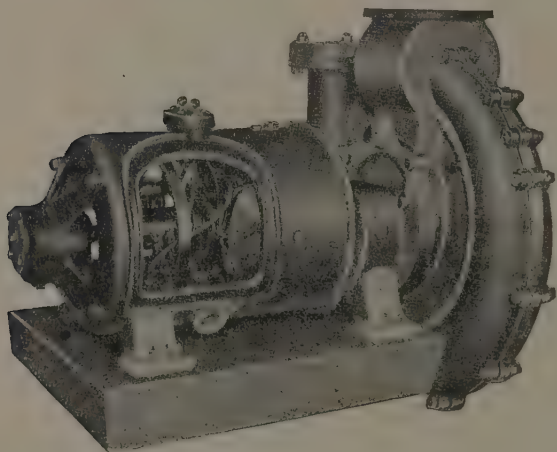


Fig. 45. Exhauster Set driven by Direct Current Motor.

As the hardest wear comes on the elbows, a patented split elbow is used, having an interchangeable wearing back, about 3" thick, made of hard iron. These wearing backs will last from 10 to 18 months, and are quickly replaced when worn out and can be replaced without interfering with the working of the conveyor, and at

trifling cost. Patented fittings with interchangeable wearing back are also used when required.

Owing to the higher velocity of the air in the central portion of the pipe, the tendency is to convey the material in suspension in the center of the pipe. That this is a fact can be readily seen on looking into a conveyor pipe in operation.

This fact prevents serious wear on the pipe, and from observation of plants in use up to two and one-half years, shows that an ordinary steel pipe, handling ashes, will last for years.

No material comes in contact with any moving part of this conveyor and it is dustless in operation. These facts should appeal to engineers, especially those who have had experience with mechanical conveyors in handling ashes.

There is absolutely no corrosion of the conveyor pipe, as the great rush of air through the pipe keeps same perfectly dry under all conditions.

The following table gives capacities, etc.:

Size of Conveyor.	Capacity per Min.
6"	200 lbs.
8"	300 lbs.
10"	500 lbs.

Firing. No definite set of rules for hand firing can be laid down that will be suitable for all steam plants, or for the many different kinds of coal used. Some kinds of coal need very little stirring or slicing, while others that have a tendency to coke, and form a crust on top of the fire need to be sliced quite often.

Every engineer, if he is at all observant, should be able to judge for himself as to the best method of treating the coal he is using, so as to get the most economical results. A few general maxims may be laid down. First, keep a clean fire; second, see that every square inch of grate surface is covered with a good live fire; third, keep a level fire, don't allow hills and valleys and yawning chasms to form in the furnace, but keep the fire level; fourth, when cleaning the fire always be sure to clean all the clinkers and dead ashes away from the back end of the grates at the bridge wall, in order that the air may have a free passage through the grate bars, because this is one of the best points in the furnace for securing good combustion, provided the bridge wall is kept clean from the grates up.

By keeping the back ends of the grate bars and the face of the bridge wall clean, the air is permitted to come in contact with the hot fire brick, and thus one of the greatest aids to good combustion is utilized. Don't allow the fire to become so deep and heavy that the air cannot pass up through it, because without a good supply of air good combustion is impossible. When the chimney draft is good the quantity of cold air admitted underneath the grate bars may be easily regulated by leaving the ash-pit doors partly open. The amount of opening required can be ascertained by a little experimenting and depends upon the intensity of the draft and the condition of the fire. With a clean, light fire, and the air spaces in the grates free from dead ashes, a slight opening of the ash-pit doors will suffice to admit all the air required beneath the grates. But if the fire is heavy and the grates are clogged, a larger opening will be necessary. In firing bituminous coal

containing a large percentage of volatile (light or gaseous matter), the best results can be obtained by leaving the fire doors slightly open for a few seconds immediately after throwing in a fresh fire. The reason for doing this is that the volatile matter in the coal flashes into flame the instant it comes in contact with the heat of the furnace, and if a sufficient supply of oxygen is not present just at this particular time the combustion will be imperfect, and the result will be the formation of carbon monoxide or carbonic oxide gas, and the loss of about two-thirds of the heat units contained in the coal. This loss can be guarded against in a great measure by a sufficient volume of air, either through the fire doors directly after putting in a fresh fire, or what is still better, providing air ducts through the bridge wall or side walls which will bring the air in on top of the fire. Each pound of coal requires for its complete combustion 12 lbs., or about 150 cu. ft., of air, and the largest volume of air is needed just after fresh coal has been added to the fire.

Cleanliness. In order to get the best results, great care should be taken that the tubes be kept clean and free from soot. Especially does this apply to horizontal return tubular boilers, for the reason that when the tubes become clogged with soot the efficiency of the draft is destroyed, and the steaming capacity of the boiler is greatly reduced. Soot not only stops the draft, but it is a non-conductor of heat. In some batteries of boilers where an inferior grade of coal is used and the draft is poor, it is absolutely necessary to scrape or blow the tubes at least once a day in order to enable the boilers to generate sufficient steam.

As to the process of cleaning there are various devices

on the market, both for blowing the soot out by means of a steam jet and also for scraping the inside of the tubes. The steam jet, if properly made and used with a high pressure and dry steam, does very satisfactory work, but it should not be depended upon exclusively to keep the tubes clean, because in process of time a scale will form inside the tubes that nothing but a good scraper will remove. For that reason it is good practice to use the scraper two or three times a week at least. When the boiler is cooled down for washing out, the bottom of the shell should be cleaned of all accumulations of dust and ashes, the combustion chamber, back of the bridge wall cleaned out, and the back flue sheet or head swept off and examined, and if there is a fusible plug in the back head the scale should be scraped from it, both inside and outside the boiler, because if it is covered with scale, neither the water nor the heat can come in contact with it, and it will be non-effective.

The following rules are compiled from those issued by various boiler insurance companies in this country and Europe—they apply to all boilers except as otherwise noted:

1. *Safety Valves.* Great care should be exercised to see that these valves are ample in size and in working order. *Overloading* or *neglect* frequently leads to the most disastrous results. Safety valves should be tried at least once every day to see that they will act freely.

2. *Pressure Gauge.* The steam gauge should stand at zero when the pressure is off, and it should show the same pressure as the safety valve when that is blowing off. If not, then one is wrong, and the gauge should be tested by one known to be correct.

3. *Water Level.* The first duty of an engineer before starting, or at the beginning of his watch, is to see that the water is at the proper height. Do not rely on glass gauges, floats or water alarms, but try the gauge cocks. If they do not agree with water gauge, learn the cause and correct it. Water level in Babcock & Wilcox boilers should be at center of drum, which is usually at middle gauge. It should not be carried above.

4. *Gauge Cocks and Water Gauges* must be kept clean. Water gauge should be blown out frequently, and the glasses and passages to gauge kept clean. The Manchester, England, Boiler Association attributes more accidents to inattention to water gauges than to all other causes put together.

5. *Feed Pump or Injector.* These should be kept in perfect order, and be of ample size. No make of pump can be expected to be continuously reliable without regular and careful attention. It is always safe to have two means of feeding a boiler. Check valves and self-acting feed valves should be frequently examined and cleaned. Satisfy yourself frequently that the valve is acting when the feed pump is at work.

6. *Low Water.* In case of low water, immediately cover the fire with ashes (wet if possible) or any earth that may be at hand. If nothing else is handy use fresh coal. Draw fire as soon as it can be done without increasing the heat. Neither turn on the feed, start or stop engine, nor lift safety valve until fires are out, and the boiler cooled down.

7. *Blisters and Cracks.* These are liable to occur in the best plate iron. When the first indication appears there must be no delay in having it carefully examined and properly cared for.

8. *Fusible Plugs*, when used, must be examined when the boiler is cleaned, and carefully scraped clean on both the water and fire sides, or they are liable not to act.

ATTENTION NECESSARY TO SECURE ECONOMY.

9. *Firing*. Fire evenly and regularly, a little at a time. Moderately thick fires are most economical, but thin firing must be used where the draught is poor. Take care to keep grates evenly covered, and allow no air-holes in the fire. Do not "clean" fires oftener than necessary. With bituminous coal, a "coking fire," *i. e.*, firing in front, and shoving back when coked, gives best results, if properly managed.

10. *Cleaning*. All heating surfaces must be kept clean outside and in, or there will be a serious waste of fuel. The frequency of cleaning will depend on the nature of fuel and water. As a rule, never allow over 1/16 inch scale or soot to collect on surfaces between cleanings. Hand-holes should be frequently removed, and surfaces examined, particularly in case of a new boiler, until proper intervals have been established by experience.

Water tube boilers are provided with extra facilities for cleaning, and with a little care can be kept up to their maximum efficiency, where tubulars or locomotive boilers would be quickly destroyed. For inspection, remove the hand-holes at both ends of the tubes, and by holding a lamp at one end and looking in at the other, the condition of the surface can be fully seen. Push the scraper through the tube to remove sediment, or if the scale is hard use the chipping scraper made for that purpose. Water through a hose will facilitate the

operation. In replacing hand-hole caps, clean the surfaces without scratching or bruising, smear with oil, and screw up tight. Examine mud-drum and remove the sediment therefrom.

The *exterior* of tubes can be kept clean by the use of blowing pipe and hose through openings provided for that purpose. In using smoky fuel, it is best to occasionally brush the surfaces when steam is off.

11. *Hot Feed-Water.* Cold water should never be fed into any boiler when it can be avoided, but when necessary it should be caused to mix with the heated water before coming in contact with any portion of the boiler.

12. *Foaming.* When foaming occurs in a boiler, checking the outflow of steam will usually stop it. If caused by dirty water, blowing down and pumping up will generally cure it. In cases of violent foaming, check the draft and fires.

13. *Air Leaks.* Be sure that all openings for admission of air to boiler or flues, except through the fire, are carefully stopped. This is frequently an unsuspected cause of serious waste.

14. *Blowing Off.* If feed-water is muddy or salt, blow off a portion frequently, according to condition of water. Empty the boiler every week or two, and fill up afresh. When surface blow-cocks are used, they should be often opened for a few minutes at a time. Make sure no water is escaping from the blow-off cock when it is supposed to be closed. Blow-off cocks and check-valves should be examined every time the boiler is cleaned.

ATTENTION NECESSARY TO SECURE DURABILITY.

15. *Leaks.* When leaks are discovered, they should be repaired as soon as possible.

16. *Blowing Off.* Never empty the boiler while the brickwork is hot.

17. *Filling Up.* Never pump cold water into a hot boiler. Many times leaks, and, in shell boilers, serious weaknesses, and sometimes explosions are the result of such an action.

18. *Dampness.* Take care that no water comes in contact with the exterior of the boiler from any cause, as it tends to corrode and weaken the boiler. Beware of all dampness in seatings or coverings.

19. *Galvanic Action.* Examine frequently parts in contact with copper or brass, where water is present, for signs of corrosion. If water is salt or acid, some metallic zinc placed in the boiler will usually prevent corrosion, but it will need attention and renewal from time to time.

20. *Rapid Firing.* In boilers with thick plates, or seams exposed to the fire, steam should be raised slowly, and rapid or intense firing avoided. With thin water tubes, however, and adequate water circulation, no damage can come from that cause.

21. *Standing Unused.* If a boiler is not required for some time, empty and dry it thoroughly. If this is impracticable, fill it quite full of water, and put in a quantity of common washing soda. External parts exposed to dampness should receive a coating of linseed oil.

22. *General Cleanliness.* All things about the boiler room should be kept clean and in good order. Negligence tends to waste and decay.

WASHING OUT.

Too much care cannot be exercised in the washing out and cleaning of boilers. The following additional remarks upon this important subject, while referring more particularly to the horizontal return tubular type, will also apply to the water tube boiler:

The length of time that a boiler can be run safely and economically after having been washed out depends upon the nature of the feed water. If the water is impregnated to a considerable extent with scale forming matter, the boiler should be washed out every two weeks at least, and in some cases of particularly bad water it becomes necessary to shorten the time to one week. To prepare a boiler for washing the fire should be allowed to burn as low as possible and then be pulled out of the furnace, the furnace doors left slightly ajar and the damper left wide open, in order that the walls may gradually cool. It is as bad a practice to cool a boiler off too suddenly as it is to fire it up too quick, because the sudden change of temperature either way has an injurious effect on the seams, contracting or expanding the plates, according as it is cooled or warmed, and thus creating leaks and very often small cracks radiating from the rivet holes, and becoming larger with each change of temperature, until finally the strength of the seam is destroyed and rupture takes place. After the boiler has become comparatively cool, and there is no pressure indicated by the steam gauge, the blow off cock may be opened and the water allowed to run out. The gauge cocks and also the drip to the water column should be left open to allow the air to enter and displace the

water. Otherwise there will be a partial vacuum formed in the boiler and the water will not run out freely.

A boiler should not be blown out (that is, emptied of water) while under pressure. The sudden change of temperature is sure to have a bad effect upon the sheets and seams. Suppose for instance that all the water is blown out of a boiler under a pressure of 20 lbs. by the steam gauge. The temperature of steam at 20 lbs. is 260° F., and it may be assumed that the metal of the boiler is at or near that temperature also. Assume the temperature of the atmosphere in the boiler-room to be 60° F. There will then be a range of $260^{\circ}-60^{\circ}=200^{\circ}$ temperature for the boiler to pass through within a short time, which will certainly have a bad effect, and besides this the boiler shell will be so hot that the loose mud and sediment left after the water has run out is liable to be baked upon the sheets, making it much harder to remove.

While inside the boiler the boiler washer should closely examine all the braces and stays, and if any are found loose or broken they should be repaired at once before the boiler is used again. The soundness of braces, rivets, etc., can be ascertained by tapping them with a light hammer.

FAILURES OF TUBES.

All tube failures reduce to four classes:

- (1) Pitting, which causes pin holes to be formed.
- (2) Defective welds, which cause the tube to open, as in A, Fig. 46.
- (3) An initial bagging resulting in a rupture, as in B.
- (4) Scabbing and blistering, as in C.

In the first case the tube is not enlarged, and may be drawn through a tube sheet, without disturbing other tubes, though usually with difficulty, owing to deposits on the outer surface.

In the other cases, the tubes become larger than their original size, hence they cannot be drawn through the tube sheet, water-leg or header, unless they are split and collapsed inch by inch for their entire length beyond the



Fig. 46. Photographs Showing Distention of Tubes at Point of Rupture.

point of failure, and if they also pass through cross baffles the enlargement will pull out the bricks and destroy the baffle. To remove a tube in this way is the work of days, and in consequence the actual method used is to cut out all tubes—numbering at times half a dozen below the defective one—and to avoid destroying the baffles these tubes are cut into several pieces.

Renewing Tubes. As it is practically impossible to prevent scale from forming on the outside of the tubes of horizontal tubular boilers unless the feed water is exceptionally good, and as the tubes will in course of time become leaky where they are expanded into the heads, the engineer, if he has a battery of two or more, should take advantage of the first opportunity that presents itself to take out of service the boiler that shows the most signs of deterioration and take out the tubes, and after cleaning them of scale by scraping and hammering or rolling in a tumbling cylinder, he should select those that are still in good condition and have them pieced out at the ends, making them almost as good as new.

The flues being out of the boiler will give the boiler washer a good opportunity to thoroughly clean the inside also, and if there are any loose rivets they should be replaced and leaky or suspicious looking seams chipped and caulked. If there are indications of corrosion or pitting, a stiff paste or putty made of plumbago mixed with a small portion of cylinder oil may be applied to the affected parts with good results.

Feed Water. There is no steam plant of any consequence that does not have more or less exhaust steam or returns from a steam heating system which can be utilized for heating the feed water before it enters the boiler. Cold water should never be pumped into a boiler that is under steam when it is possible to prevent it.

FEED WATER HEATERS.

Feed Water Heaters. One great source of economy in fuel is the utilization of all the available exhaust steam for heating the feed water before it enters the boiler. Of course if the main engines are condensing, the exhaust from that source is not directly available, except by interposing a closed heater between the cylinder and the condenser, or by using the water of condensation for feeding the boilers. This can be done with safety, provided a surface condenser is used, but with a jet condenser or an open heater in which the exhaust mingles with the water, it is advisable to have an oil separator to prevent the oil from getting into the boilers.

Exhaust heaters are of two kinds, open and closed. In the open heater the exhaust steam mingles directly with the water and a portion of it is condensed. A well-designed open exhaust heater will raise the temperature of the water to very nearly the boiling point, 212° F. These heaters should be set so that the water will flow by gravity from them to the feed pump. In the closed type of exhaust heaters the exhaust steam and the water are kept separate. In some styles the steam passes through tubes, which are surrounded by water, while in others the water fills the tubes, which are in turn surrounded by the steam. In either case the water in the closed heater is under the full boiler pressure, while the feed pump is in operation, because the heater is between the pump and the boiler, while with the open heater the pump is between the heater and the boiler.

The saving effected by heating the feed water with exhaust steam can be easily ascertained by the use of a thermometer, a steam table, and a simple arithmetical calculation. First, find by thermometer the temperature



Fig. 47. Baragwanath Steam Jacket Feed Water Heater.

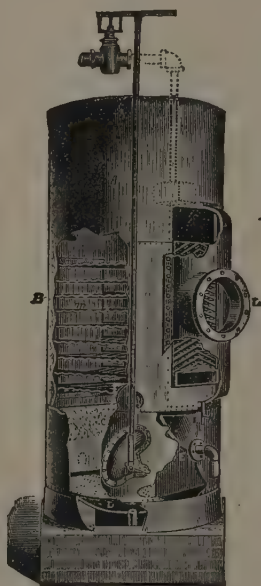


Fig. 48. Interior View of Open Heater.

of the water before entering the heater; find its temperature as it leaves the heater. Next ascertain by table 10 the number of heat units above 32° F. in the water at each of the two temperatures. Subtract the less from the greater, and the remainder will be the

number of heat units added to the water by the heater. Next find by the table the number of heat units above 32° F. in the steam at the pressure ordinarily carried in the boiler, and subtract from this the number of heat units in the water before it enters the heater. The result will be the number of heat units that would be required to convert the water into steam of the required pressure, provided no heater were used. Then to find the percentage of saving effected by the heater, multiply the number of heat units added to the water by the heater by 100, and divide by the number of heat units required to convert the unheated water into steam, from the initial temperature at which it enters the heater.

Example. Assume the boiler to be carrying 100 lbs. gauge pressure. Suppose the temperature of the water before entering the heater is 60° F., and that after leaving the heater its temperature is 202° F., what is the percentage of saving due to the heater? The solution of the problem is as follows:

Boiler pressure by gauge = 100 lbs.

Initial temperature of feed water = 60° F.

Heated temperature of feed water = 202° F.

From Table 10 it is found that

Heat units in water at 202° F. = 170.7.

Heat units in water at 60° F. = 28.01.

Heat units added to water by heater = $170.7 - 28.01$
= 142.69.

Heat units in steam at 100 lbs. gauge pressure = 1185.0.

Heat units to be added to water at 60° F. to make steam of 100 lbs. gauge pressure = $1185.0 - 28.01$ = 1156.99.

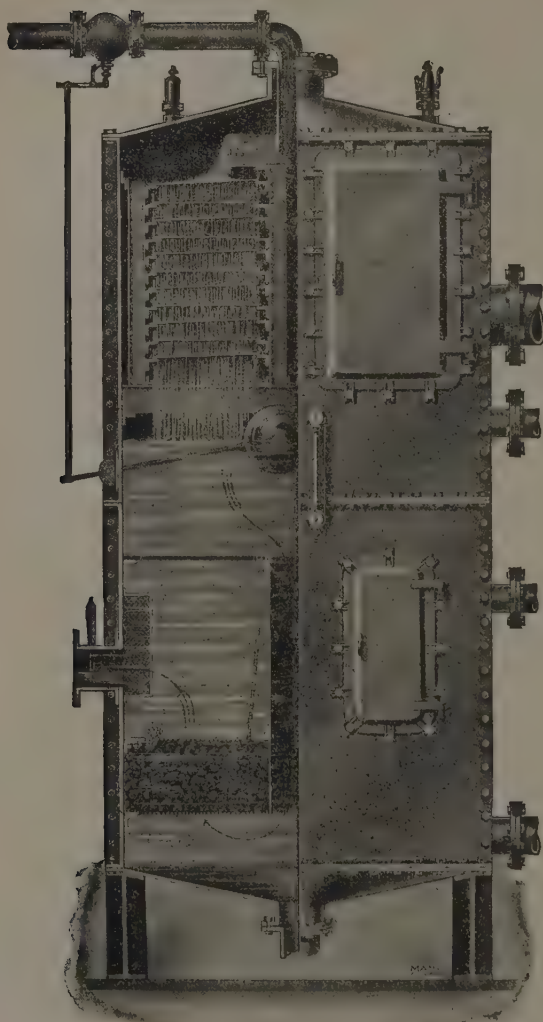


Fig. 49. Square Open Heater.

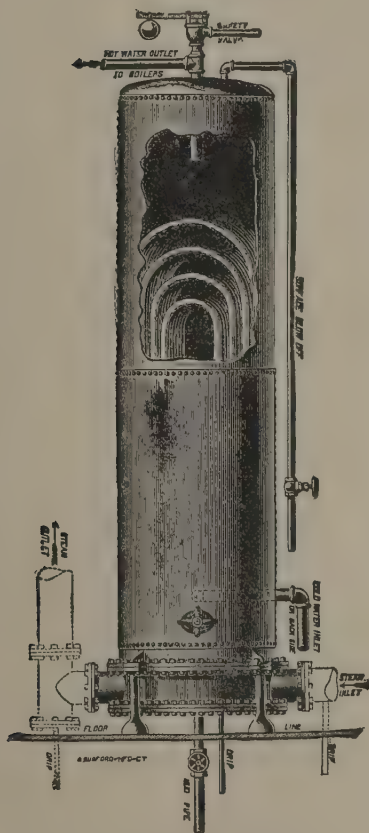


Fig. 50. Closed Feed Water Heater.

Percentage of saving effected by the use of the heater

$$= \frac{142.69 \times 100}{1156.99} = 12.33 \text{ per cent.}$$

Suppose the coal consumed under this boiler amounts to two tons per day at a cost of \$3.00 per ton, or a fuel cost of \$6.00 per day. Then the saving in dollars and cents due to the heater in the foregoing example would be 12.33 per cent of \$6.00, or \$0.7398 (74 cents) per day.

TABLE 10

PROPERTIES OF SATURATED STEAM

Vacuum Inches of Mercury	Absolute Pressure Lbs. per Sq. Inch	Temp. Degrees F.	Total Heat above 32° F.		Latent Heat H-h Heat units	Relative Volume	Cubic Feet in 1 Lb. Wt. of Steam	Wt. of 1 Cubic Foot of Steam, Lbs.
			In the Water h Heat-units	In the Steam H Heat-units				
29.74	.089	32.	0.	1091.7	1091.7	208,080	3333.3	.0003
29.67	.122	40.	8.	1094.1	1086.1	154,330	2472.2	.0004
29.56	.176	50	18.	1097.2	1079.2	107,630	1724.1	.0006
29.40	.254	60.	28.01	1100.2	1072.2	76,370	1223.4	.0008
29.19	.359	70.	38.02	1103.3	1065.3	54,660	875.61	.0011
28.90	.502	80.	48.04	1106.3	1058.3	39,690	635.80	.0016
28.51	.692	90.	58.06	1109.4	1051.3	29,290	469.20	.0021
28.00	.943	100.	68.08	1112.4	1044.4	21,830	349.70	.0028
27.88	1.	102.1	70.09	1113.1	1043.0	20,623	334.23	.0030
25.85	2.	126.3	94.44	1120.5	1026.0	10,730	173.23	.0058
23.83	3.	141.6	109.9	1125.1	1015.3	7,325	118.00	.0085
21.78	4.	153.1	121.4	1128.6	1007.2	5,588	89.80	.0111
19.74	5.	162.3	130.7	1131.4	1000.7	4,530	72.50	.0127
17.70	6.	170.1	138.6	1133.8	995.2	3,816	61.10	.0163
15.67	7.	176.9	145.4	1135.9	990.5	3,302	53.00	.0189
13.63	8.	182.9	151.5	1137.7	986.2	2,912	46.60	.0214
11.60	9.	188.3	156.9	1139.4	982.4	2,607	41.82	.0239
9.56	10.	193.2	161.9	1140.9	979.0	2,361	37.80	.0264
7.52	11.	197.8	166.5	1142.3	975.8	2,159	34.61	.0289
5.49	12.	202.0	170.7	1143.5	972.8	1,990	31.90	.0314
3.45	13.	205.9	174.7	1144.7	970.0	1,846	29.60	.0338
1.41	14.	209.6	178.4	1145.9	967.4	1,721	27.50	.0363
0.00	14.7	212.0	180.9	1146.6	963.7	1,616	26.26	.0379

TABLE 10—Continued

Gauge Pressure Lbs. per Sq. In.	Absolute Pressure Lbs. per Sq. In.	Temp. Degrees F.	Total Heat Above 32° F.		Latent Heat H _h Heat-units	Relative Volume	Cubic Feet in 1 Lb. Wt. of Steam	Wt. of 1 Cubic Foot of Steam, Lbs.
			In the Water H Heat-units	In the Steam H Heat-units				
0.3	15	213.3	181.9	1146.9	965.0	1.614	25.90	.0387
1.3	16	216.3	185.3	1147.9	962.7	1.519	24.33	.0411
2.3	17	219.4	188.4	1148.9	960.5	1.434	23.00	.0435
3.3	18	222.4	191.4	1149.8	958.3	1.359	21.80	.0459
4.3	19	225.2	194.3	1150.6	956.3	1.292	20.70	.0483
5.3	20	227.9	197.0	1151.5	954.4	1.231	19.72	.0507
6.3	21	230.5	199.7	1152.2	952.6	1.176	18.84	.0531
7.3	22	233.0	202.2	1153.0	950.8	1.126	18.03	.0555
8.3	23	235.4	204.7	1153.7	949.1	1.080	17.30	.0578
9.3	24	237.8	207.0	1154.5	947.4	1.038	16.62	.0602
10.3	25	240.0	209.3	1155.1	945.8	998	16.00	.0625
11.3	26	242.2	211.5	1155.8	944.3	962	15.42	.0649
12.3	27	244.3	213.7	1156.4	942.8	929	14.90	.0672
13.3	28	246.3	215.7	1157.1	941.3	898	14.40	.0696
14.3	29	248.3	217.8	1157.7	939.9	869	13.91	.0719
15.3	30	250.2	219.7	1158.3	938.9	841	13.50	.0742
16.3	31	252.1	221.6	1158.8	937.2	816	13.07	.0765
17.3	32	254.0	223.5	1159.4	935.9	792	12.68	.0788
18.3	33	255.7	225.3	1159.9	934.6	769	12.32	.0812
19.3	34	257.5	227.1	1160.5	933.4	748	12.00	.0835
20.3	35	259.2	228.8	1161.0	932.2	728	11.66	.0858
21.3	36	260.8	230.5	1161.5	931.0	709	11.36	.0880
22.3	37	262.5	232.1	1162.0	929.8	691	11.07	.0903
23.3	38	264.0	233.8	1162.5	928.7	674	10.80	.0926
24.3	39	265.6	235.4	1162.9	927.6	658	10.53	.0949
25.3	40	267.1	236.9	1163.4	926.5	642	10.28	.0972
26.3	41	268.6	238.5	1163.9	925.4	627	10.05	.0995
27.3	42	270.1	240.0	1164.3	924.4	613	9.83	.1018
28.3	43	271.5	241.4	1164.7	923.3	600	9.61	.1040
29.3	44	272.9	242.9	1165.2	922.3	587	9.41	.1063
30.3	45	274.3	244.3	1165.6	921.3	575	9.21	.1086
31.3	46	275.7	245.7	1166.0	920.4	563	9.02	.1108
32.3	47	277.0	247.0	1166.4	919.4	552	8.84	.1131
33.3	48	278.3	248.4	1166.8	918.5	541	8.67	.1153
34.3	49	279.6	249.7	1167.2	917.5	531	8.50	.1176
35.3	50	280.9	251.0	1167.6	916.6	520	8.34	.1198
36.3	51	282.1	252.2	1168.0	915.7	511	8.19	.1221
37.3	52	283.3	253.5	1168.4	914.9	502	8.04	.1243

TABLE 10—Continued

Gauge Pressure Lbs. per Sq. In.	Absolute Pressure Lbs. per Sq. In.	Temp. Degrees F.	Total Heat above 32° F.		Latent Heat H _h Heat-units	Relative Volume	Cubic Feet in 1 Lb. Wt. of Steam	Wt. of 1 Cubic Foot of Steam, Lbs.
			In the Water h Heat-units	In the Steam H Heat-units				
38.3	53	284.5	254.7	1168.7	914.0	492	7.90	.1266
39.3	54	285.7	256.0	1169.1	913.1	484	7.76	.1288
40.3	55	286.9	257.2	1169.4	912.3	476	7.63	.1311
41.3	56	288.1	258.3	1169.8	911.5	468	7.50	.1333
42.3	57	289.1	259.5	1170.1	910.6	460	7.38	.1355
43.3	58	290.3	260.7	1170.5	909.8	453	7.26	.1377
44.3	59	291.4	261.8	1170.8	909.0	446	7.14	.1400
45.3	60	292.5	262.9	1171.2	908.2	439	7.03	.1422
46.3	61	293.6	264.0	1171.5	907.5	432	6.92	.1444
47.3	62	294.7	265.1	1171.8	906.7	425	6.82	.1466
48.3	63	295.7	266.2	1172.1	905.9	419	6.72	.1488
49.3	64	296.8	267.2	1172.4	905.2	413	6.62	.1511
50.3	65	297.8	268.3	1172.8	904.5	407	6.53	.1533
51.3	66	298.8	269.3	1173.1	903.7	401	6.43	.1555
52.3	67	299.8	270.4	1173.4	903.0	395	6.34	.1577
53.3	68	300.8	271.4	1173.7	902.3	390	6.25	.1599
54.3	69	301.8	272.4	1174.0	901.6	384	6.17	.1621
55.3	70	302.7	273.4	1174.3	900.9	379	6.09	.1643
56.3	71	303.7	274.4	1174.6	900.2	374	6.01	.1665
57.3	72	304.6	275.3	1174.8	899.5	369	5.93	.1687
58.3	73	305.6	276.3	1175.1	898.9	365	5.85	.1709
59.3	74	306.5	277.2	1175.4	898.2	360	5.78	.1731
60.3	75	307.4	278.2	1175.7	897.5	356	5.71	.1753
61.3	76	308.3	279.1	1176.0	896.9	351	5.63	.1775
62.3	77	309.2	280.0	1176.2	896.2	347	5.57	.1797
63.3	78	310.1	280.9	1176.5	895.6	343	5.50	.1819
64.3	79	310.9	281.8	1176.8	895.0	339	5.43	.1840
65.3	80	311.8	282.7	1177.0	894.3	334	5.37	.1862
66.3	81	312.7	283.6	1177.3	893.7	331	5.31	.1884
67.3	82	313.5	284.5	1177.6	893.1	327	5.25	.1906
68.3	83	314.4	285.3	1177.8	892.5	323	5.18	.1928
69.3	84	315.2	286.2	1178.1	891.9	320	5.13	.1950
70.3	85	316.0	287.0	1178.3	891.3	316	5.07	.1971
71.3	86	316.8	287.9	1178.6	890.7	313	5.02	.1993
72.3	87	317.7	288.7	1178.8	890.1	309	4.96	.2015
73.3	88	318.5	289.5	1179.1	889.5	306	4.91	.2036
74.3	89	319.3	290.4	1179.3	888.9	303	4.86	.2058
75.3	90	320.0	291.2	1179.6	888.4	299	4.81	.2080

TABLE 10—*Continued*

Gauge Pressure Lbs. per Sq. In.	Absolute Pressure Lbs. per Sq. In.	Temp. Degrees F.	Total Heat above 32° F.		Latent Heat H-h Heat-units	Relative Volume	Cubic Feet in 1 Lb. Wt. of Steam	Wt. of 1 Cubic Foot of Steam, Lbs.
			In the Water h Heat-units	In the Steam H Heat-units				
76.3	91	320.8	292.0	1179.8	887.8	296	4.76	.2102
77.3	92	321.6	292.8	1180.0	887.2	293	4.71	.2123
78.3	93	322.4	293.6	1180.3	886.7	290	4.66	.2145
79.3	94	323.1	294.4	1180.5	886.1	287	4.62	.2166
80.3	95	323.9	295.1	1180.7	885.6	285	4.57	.2188
81.3	96	324.6	295.9	1181.0	885.0	282	4.53	.2210
82.3	97	325.4	296.7	1181.2	884.5	279	4.48	.2231
83.3	98	326.1	297.4	1181.4	884.0	276	4.44	.2253
84.3	99	326.8	298.2	1181.6	883.4	274	4.40	.2274
85.3	100	327.6	298.9	1181.8	882.9	271	4.36	.2296
86.3	101	328.3	299.7	1182.1	882.4	268	4.32	.2317
87.3	102	329.0	300.4	1182.3	881.9	266	4.28	.2339
88.3	103	329.7	301.1	1182.5	881.4	264	4.24	.2360
89.3	104	330.4	301.9	1182.7	880.8	261	4.20	.2382
90.3	105	331.1	302.6	1182.9	880.3	259	4.16	.2403
91.3	106	331.8	303.3	1183.1	879.8	257	4.12	.2425
92.3	107	332.5	304.0	1183.4	879.3	254	4.09	.2446
93.3	108	333.2	304.7	1183.6	878.8	252	4.05	.2467
94.3	109	333.9	305.4	1183.8	878.3	250	4.02	.2489
95.3	110	334.5	306.1	1184.0	877.9	248	3.98	.2510
96.3	111	335.2	306.8	1184.2	877.4	246	3.95	.2531
97.3	112	335.9	307.5	1184.4	876.9	244	3.92	.2553
98.3	113	336.5	308.2	1184.6	876.4	242	3.88	.2574
99.3	114	337.2	308.8	1184.8	875.9	240	3.85	.2596
100.3	115	337.8	309.5	1185.0	875.5	238	3.82	.2617
101.3	116	338.5	310.2	1185.2	875.0	236	3.79	.2638
102.3	117	339.1	310.8	1185.4	874.5	234	3.76	.2660
103.3	118	339.7	311.5	1185.6	874.1	232	3.73	.2681
104.3	119	340.4	312.1	1185.8	873.6	230	3.70	.2703
105.3	120	341.0	312.8	1185.9	873.2	228	3.67	.2764
106.3	121	341.6	313.4	1186.1	872.7	227	3.64	.2745
107.3	122	342.2	314.1	1186.3	872.3	225	3.62	.2766
108.3	123	342.9	314.7	1186.5	871.8	223	3.59	.2788
109.3	124	343.5	315.3	1186.7	871.4	221	3.56	.2809
110.3	125	344.1	316.0	1186.9	870.9	220	3.53	.2830
111.3	126	344.7	316.6	1187.1	870.5	218	3.51	.2851
112.3	127	345.3	317.2	1187.3	870.0	216	3.48	.2872
113.3	128	345.9	317.8	1187.4	869.6	215	3.46	.2894

TABLE 10—Continued

Gauge Pressure Lbs. per Sq. In.	Absolute Pressure Lbs. per Sq. In.	Temp. Degrees F.	Total Heat Above 32° F.		Latent Heat H-h Heat-units	Relative Volume	Cubic Feet in 1 Lb. Wt. of Steam	Wt. of 1 Cubic Foot of Steam, Lbs.
			In the Water h Heat-units	In the Steam H Heat-units				
114.3	129	346.5	318.4	1187.6	869.2	213	3.43	.2915
115.3	130	347.1	319.1	1187.8	868.7	212	3.41	.2936
116.3	131	347.6	319.7	1188.0	868.3	210	3.38	.2957
117.3	132	348.2	320.3	1188.2	867.9	209	3.36	.2978
118.3	133	348.8	320.8	1188.3	867.5	207	3.33	.3000
119.3	134	349.4	321.5	1188.5	867.0	206	3.31	.3021
120.3	135	350.0	322.1	1188.7	866.6	204	3.29	.3042
121.3	136	350.5	322.6	1188.9	866.2	203	3.27	.3063
122.3	137	351.1	323.2	1189.0	865.8	201	3.24	.3084
123.3	138	351.8	323.8	1189.2	865.4	200	3.22	.3105
124.3	139	352.2	324.4	1189.4	865.0	199	3.20	.3126
125.3	140	352.8	325.0	1189.5	864.6	197	3.18	.3147
126.3	141	353.3	325.5	1189.7	864.2	196	3.16	.3169
127.3	142	353.9	326.1	1189.9	863.8	195	3.14	.3190
128.3	143	354.4	326.7	1190.0	863.4	193	3.11	.3211
129.3	144	355.0	327.2	1190.2	863.0	192	3.09	.3232
130.3	145	355.5	327.8	1190.4	862.6	191	3.07	.3253
131.3	146	356.0	328.4	1190.5	862.2	190	3.05	.3274
133.3	148	357.1	329.5	1190.9	861.4	187	3.02	.3316
135.3	150	358.2	330.6	1191.2	860.6	185	2.98	.3358
140.3	155	360.7	333.2	1192.0	858.7	179	2.89	.3463
145.3	160	363.3	335.9	1192.7	856.9	174	2.80	.3567
150.3	165	365.7	338.4	1193.5	855.1	169	2.72	.3671
155.3	170	368.2	340.9	1194.2	853.3	164	2.65	.3775
160.3	175	370.5	343.4	1194.9	851.6	160	2.58	.3879
165.3	180	372.8	345.8	1195.7	849.9	156	2.51	.3983
170.3	185	375.1	348.1	1196.3	848.2	152	2.45	.4087
175.3	190	377.3	350.4	1197.0	846.6	148	2.39	.4191
180.3	195	379.5	352.7	1197.7	845.0	144	2.33	.4296
185.3	200	381.6	354.9	1198.3	843.4	141	2.27	.4400
190.3	205	383.7	357.1	1199.0	841.9	138	2.22	.4503
195.3	210	385.7	359.2	1199.6	840.4	135	2.17	.4605
200.3	215	387.7	361.3	1200.2	838.9	132	2.12	.4707
205.3	220	389.7	362.2	1200.8	838.6	129	2.06	.4852
245.3	260	404.4	377.4	1205.3	827.9	110	1.76	.5686
285.3	300	417.4	390.9	1209.2	818.3	96	1.53	.6515
485.3	500	467.4	443.5	1224.5	781.0	59	.94	1.062
685.3	700	504.1	482.4	1235.7	753.3	42	.68	1.470
985.3	1000	546.8	528.3	1248.7	720.3	30	.48	2.082

THE HOPPES CLASS R FEED-WATER HEATER.

Figure 51 herewith is a side sectional elevation of a Hoppes class R open feed-water heater, Fig. 52 being an end sectional view of the same. Although this heater has been on the market several years, it has recently been improved and embodies features not heretofore shown. The shell of the heater is cylindrical and the heads are "bumped," a design calculated to resist pressure, and

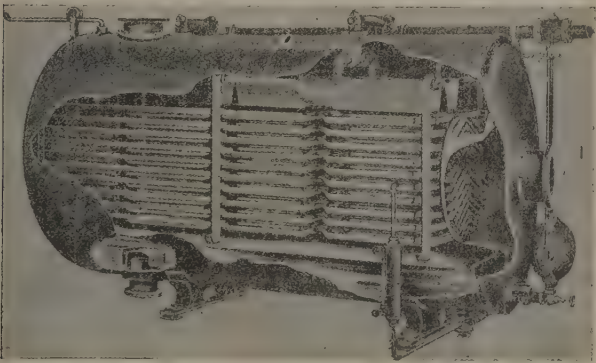


Fig. 51. Sectional Elevation of Hoppes Class R Feed-Water Heater.

also to prevent pulsations due to the impulses of the exhaust. The interior of the heater is provided with layers of trough-shaped pans arranged in tiers and designed to afford a large amount of heating surface. To avoid corrosion the pans are of cast-iron, as are also the bottom of the shell, the lower ends of the center posts and all other parts with which the water comes in contact. The shell may be entirely cast-iron, however, if desired. In the back end of the heater is located a large oil catcher, through which the exhaust passes. See Fig. 51.

The principle of operation is to provide that the flow of water from the cast-iron troughs at the top be so gradual that the water will be distributed over the edges of the pans in a thin film, and flow over the sides and ends so gently that it will follow the bottom contour of each pan to the lowest point before dropping off into the pan beneath. While the water is following the under

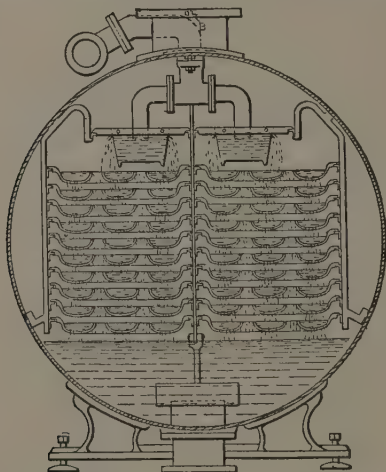


Fig. 52. End Sectional View of Hoppes Class R Heater.

side of the pans, the exhaust steam will come in direct contact with and heat it to the temperature of the exhaust. Lime and other solids which may be held in solution in the water will, when liberated by the heat, form mostly on the under side of the pans and hence will not detract from the efficiency of the process, as the same direct action of the exhaust on the water will continue as when the pans are clean.

In open heaters of large size the regulation and distribution of water so as to obtain the best results are important. In the Hoppes apparatus the water is regulated to the heater by a float in a separate float-chamber operating a double-disk balanced valve of the company's manufacture. Branch pipes from the main feed-pipe are connected to the shell at the top and these branches have extensions inside the heater to which are attached flanged tees. To the flanges of these tees are bolted flanged inverted L-shaped pipes, the long arms of which extend below the water level of the feed troughs in the top of the heater. Disks with orifices of proper and equal size are placed between the flanges of the L-shaped feed-pipes and the tees, and by this means the water is equally distributed into the various tiers of pans.

When it is desired to feed two or more heaters in multiple, an equal distribution of water is given to all the heaters by using a single regulating valve on the main-water inlet, and branching the same to the heaters to be supplied.

A feature recently added is an overflow dam, at the rear end of the heater, which is intended to act as a skimmer and also to add to the capacity of the overflow pipe by increasing the head on the outlet without causing the water to rise higher in the heater. The drip from the oil separator is piped into the chamber formed by the dam, but this may have a separate connection, if preferred. Filters are provided on request, but it is believed that the large amount of lime-catching surfaces obviates the necessity for their use in most cases.

These heaters are built in sizes ranging from 50 to 30,000 horse-power.

Heaters, especially those of the closed type, should have capacity sufficient to supply the boiler for fifteen or twenty minutes. There would then be a body of water continually in the heater in direct contact with the heating surface, and as it passes slowly through it will receive much more heat than if rushed through a heater that is too small. All heaters and feed pipes should be well protected by some good insulating covering to prevent loss of heat by radiation. In some cases the exhaust steam, or a portion of it at least, can be used to advantage in an exhaust injector. This device, where it can be used at all, is economical in that it not only feeds the boiler, but also heats the water without the use of live steam. But it will not force the water against a pressure much above 75 lbs. to the square inch, and if the initial temperature of the water is much above 75° F. the exhaust injector will not handle it. Heaters which use live steam direct from the boilers heat the feed water to a much higher temperature, so that they act as purifiers by removing a large portion of the scale-forming impurities before the water enters the boiler. Live steam heaters, however, are not to be considered as economizers of heat.

Provisions for Testing. While considering feed pipes and other apparatus necessarily appertaining to the feeding of boilers, it is well to devote a short space also to the fittings and other devices required for successfully conducting tests of the boiler and furnace. This subject is mentioned here for the reason that the author considers that the necessary fittings and appliances for making evaporative tests properly belong to, and in fact are a part of, the feed piping, and can be put in while the plant is being erected at much less cost and trouble than

if the matter is postponed until after the plant is in operation.

Beginning then at the check valve, there should be a tee located in the horizontal section of the feed pipe, as near to the check valve as practicable, and between it and the feed pump: or a tee can be used in place of an ell to connect the vertical and horizontal sections of the branch pipe where it rises in front of the boiler. One opening of this tee is reduced to $\frac{3}{8}$ or $\frac{1}{2}$ in. to permit the attachment of a hot water thermometer. These thermometers are also made angle-shaped at the shank, so that if desired they can be screwed into a tee placed in vertical pipe and still allow the scale to stand vertical. The thermometer is for the purpose of showing at what temperature the feed water enters the boiler during the test, and therefore should be as near the boiler as possible. After the test is completed the thermometer may be taken out and a plug inserted in its place.

The next requirement will be a device of some kind for ascertaining the weight of water pumped into the boiler during the test. In some well ordered plants each boiler is fitted with a hot water meter in the feed pipe, but as this arrangement is hardly within the reach of all, a substitute equally as accurate can be made by placing two small water tanks, each having a capacity of eight or ten cubic feet, in the vicinity of the feed pump. These tanks can be made of light tank iron, and each should be fitted with a nipple and valve near the bottom for connection with the suction side of the pump. The tops of the tanks may be left open. If an open heater is used, and it is possible to place the tanks low enough to allow a portion of the water from the heater to be led into them by gravity, it will be desirable to do so. A

pipe leading from the main water supply, with a branch to each tank, is also needed for filling them. One of the feed pumps, of which there should always be at least two, as already stated, is fitted with a tee in the suction pipe near the pump to receive the pipe leading from the tanks. During the test the main suction leading to this pump from the general supply should be kept closed, so that only the water that passes through the tanks is used for feeding the boiler. If the plant be a small one, with but one or two boilers and only a single feed pump, the latter can be made to do duty as a testing pump, because during the test there will be no other boilers to feed besides the ones under test.

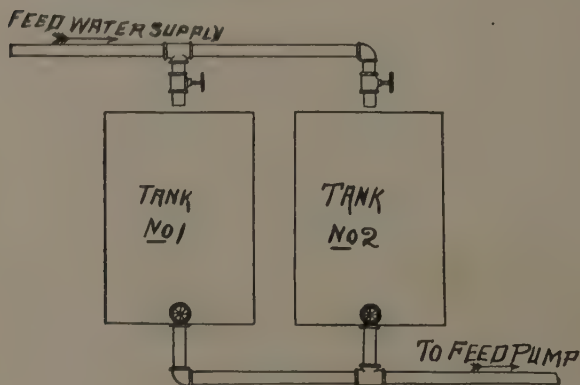


Fig. 53.

If metal tanks are considered too expensive, two good water-tight barrels can be substituted. Fig. 53 will give the reader a general idea of what is needed for obtaining the weight of the water by the method just described.

If a closed heater is used and no other boilers are in service during the test, the cold water can be measured in the tanks and pumped directly through the heater, but if it is necessary to feed other boilers besides those under test, then either a separate feed pipe must be run to the test boilers, or else hot water meters will have to be put into the branch pipes.

In cases where a separate feed pipe must be put in for the test boiler and the water which is used for testing cannot be passed through a heater, there should be a $\frac{3}{4}$ or 1 in. pipe connected to the feed main or header and leading to the testing tanks, in order to allow a portion of the hot feed water to run into and mix with the cold water in the tanks as they are being filled, thus partially warming the water before it goes to the boiler.

Heating Surface. The heating surface of a boiler consists of that portion of the boiler which is exposed to the heat on one side and water on the other. In a horizontal boiler of either the flue or tubular type, the available heating surface is, first, the lower half of the shell; second, the area of the back head below the water line minus the combined cross sectional area of all the tubes or flues; third, the inside area of the flues; fourth, the area of the front head minus the sectional area of the flues.

For a fire-box boiler of the vertical type, the area of the flue sheets, minus the sectional area of the flues, plus the area of the fire-box, plus the inside area of the flues constitutes the heating surface. If the boiler is a horizontal internally fired boiler, the heating surface will consist of, first, area of three sides of the fire-box; second, area of the crown sheet; third, area of flue sheets minus sectional area of flues; fourth, inside area of the flues.

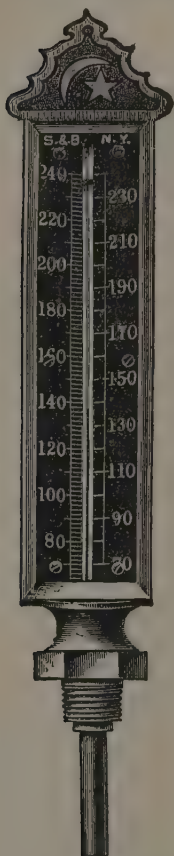


Fig. 54. Hot Water Thermometer.

eter is 15.9 sq. in. Number of flues, $62 \times 15.9 = 985.8$ sq.

In estimating the area of the fire-box, the area of the fire door should be subtracted therefrom. If the fire-box be circular, as in the case of a vertical boiler, the area may be obtained by first finding by measurements the diameter, which, multiplied by 3.1416, will give the circumference. Then multiply this result by the height or the distance between the grate bars and the flue sheet. In the case of water tube boilers the outside area of the tubes must be taken. Two examples will be given illustrating methods of calculating heating surface:

First, take a horizontal tubular boiler diameter 72 in., length 18 ft., having sixty-two $4\frac{1}{2}$ in. flues; find area of lower half of shell.

Circumference = diameter $\times$ 3.1416,
= 18.8496 ft.

One-half of the circumference multiplied by the length = required area. Thus, $18.8496 \div 2 \times 18 = 169.64$ sq. ft.

Next find heating surface of back head below the water line. Total area = $72^2 \times .7854 = 4071.5$ sq. in. Assume two-thirds of this area to be exposed to the heat. $\frac{2}{3}$ of 4071.5 = 2714.3 sq. in. From this must be deducted the sectional area of the tubes. In giving the size of boiler tubes the outside diameter is taken. The tubes being $4\frac{1}{2}$ in.; the area of a circle $4\frac{1}{2}$ in. in diameter is 15.9 sq. in. Number of flues, $62 \times 15.9 = 985.8$ sq.

in. = sectional area of tubes. The heating surface of the back head therefore = $2714.3 - 985.8 = 1728.5$ sq. in. Dividing this by 144, to reduce to feet, we have 12 sq. ft.

Next find inside area of tubes. The standard thickness of a $4\frac{1}{2}$ in. tube = .134 in. The inside diameter therefore will be $4.5 - (2 \times .134) = 4.23$ in., and the circumference will be $4.23 \times 3.1416 = 13.29$ in., and the inside area will be $13.29 \times \text{length, 18 ft.,} = 216$ in. Thus $216 \times 13.29 \div 144 = 19.93$ sq. ft., inside area of one flue. There being 62 flues, the total heating surface of tubes is $19.93 \times 62 = 1235.66$ sq. ft. The heating surface of the front head is found in the same manner as that of the back head, with the exception that the whole area should be figured instead of two-thirds, for the reason that the entire surface is exposed to the heat, although that portion above the water line may be considered as superheating surface. The heating surface of front head would be: area 4071.5 — sectional area of tubes, 985.8 = 3085.7 sq. in. = 21.43 sq. ft.

The total heating surface of the boiler is thus found to be 1438.73 sq. ft., divided up as follows:

Lower half of shell	169.64 sq. ft.
Back head,	12.00 “
Tubes,	1235.66 “
Front head,	21.43 “
	1438.73 “

Next taking a vertical fire-box boiler of the following dimensions: diameter of flue sheet, and also of fire-box, 50 in.; height of fire-box above grate bars, 30 in.; number of flues, 200; size of flues, 2 in.; length of flues, 7 ft.

First, find heating surface in flue sheet.

Area of circle, 50 in. in diameter = 1,963.5 sq. in.

Sectional area of 2 in. flue = 3.14 sq. in., which multiplied by 200 = 628 sq. in., total sectional area of tubes. The heating surface of one flue sheet therefore will be $1,963.5 - 628 \div 144 = 9$ sq. ft.

Assuming that the tops of the flues are submerged, the area of the top flue sheet will also be 9 sq. ft. Then heating surface of flue sheets = $9 \times 2 = 18$ sq. ft.

Second, find heating surface of tubes. The standard thickness of a 2 in. flue is .095 in. The inside diameter will consequently be $2 - (.095 \times 2) = 1.8$ in., and the circumference will be $1.8 \times 3.1416 = 5.66$ in. The length of the flue being 7 ft., or 84 in., the inside area will be $5.66 \times 84 \div 144 = 3.3$ sq. ft., and multiplying this result by 200 we have $200 \times 3.3 = 660$ sq. ft. as the heating surface of the flues.

Third, find heating surface of the fire-box. Diameter of fire-box = 50 in., which multiplied by 3.1416 = 157.08, which is the circumference. The height being 30 in., the total area will be $157.08 \times 30 \div 144 = 32.7$ sq. ft. Allowing 1 sq. ft. as the area of the fire door, will leave 31.7 sq. ft. heating surface of fire-box. The heating surface of the boiler will be:

For the flue sheets	18	sq. ft.
For the flues,	660	"
For the fire-box,	31.7	"
Total,	709.7	"

The above methods may be applied in estimating the heating surface of any boiler, provided in the case of

water tube boilers that the outside in place of the inside area of the tubes can be figured.

Water Columns. Water columns should be so located as to bring the lower end of the gauge glass exactly on a level with the top of the upper row of tubes, thus always affording a perfect guide as to the depth of water over the tubes. Many gauge glasses are placed too low, and water tenders and firemen are often deceived by them unless their positions with relation to the tubes are carefully noted.

The only safe plan for an engineer to pursue in taking charge of a steam plant is to seize the first opportunity for noting this relation. When he has washed out his boilers he may leave the top man-hole plates out while refilling them, and when the water stands at about four inches over the top row of tubes, the depth of water in the glass should be measured. He should do this with every boiler in the plant, and make a memorandum for each boiler. He will then know his bearings with regard to the safe height of water to be carried in the several gauge glasses. If he finds any of them are too low, he should lose no time in having them altered to conform to the requirements of safety. The position of the lower gauge cock should be three inches above the top row of tubes.

In making connections for the water column plugged crosses should always be used in place of ells. Brass plugs are to be preferred if they can be obtained; but whether of brass or iron, they should always be well coated with a paste made of graphite and cylinder oil before they are screwed in. They can then be easily removed when washing out the boiler, so as to allow the scale, which is sure to form in the lower connection, to

be cleaned out. The best point at which to connect the lower pipe with the boiler is in the lower part of the head just below the bottom row of tubes, and near the side of the boiler on which the water column is to stand; $1\frac{1}{4}$ or $1\frac{1}{2}$ in. pipe should be used in all cases. The top connection can be made either in the head near the top, or in the shell. A $\frac{3}{4}$ or 1 in. drain pipe should be led into the ash pit, fitted with a good reliable valve which should be opened at frequent intervals to allow the mud and dirt to blow out of the water column and its connections. This is a very important point, and great care should be taken to keep the water column and all its connections thoroughly clean at all times.

One of the best indications that some portion of the connections between the water glass and the boiler is choked or plugged with scale, is when there is no perceptible movement of the water in the glass. When the connections are free and the boiler is being fired, there is always a slight movement of the water up and down in the glass, and when there is no perceptible movement it is time to look for the cause at once. Many instances of burned tubes have occurred, and even explosions caused by low water in boilers while the gauge glass showed the water to be at a safe height. But owing to the connections having become plugged with scale, the water in the glass had no connection whatever with that in the boiler, and the water column was therefore worse than useless.

The above remarks apply particularly to horizontal return tubular boilers. In the case of water tube boilers the location of the gauge glass as regards height is gov-

erned by the desired level of the water in steam drum, or drums.

Safety Valves. The modern pop valve is generally reliable, but, like everything else, if it is allowed to stand idle too long it is likely to become rusty and stick. Therefore it should be allowed to blow off at least once or twice a week in order to keep it in good condition.

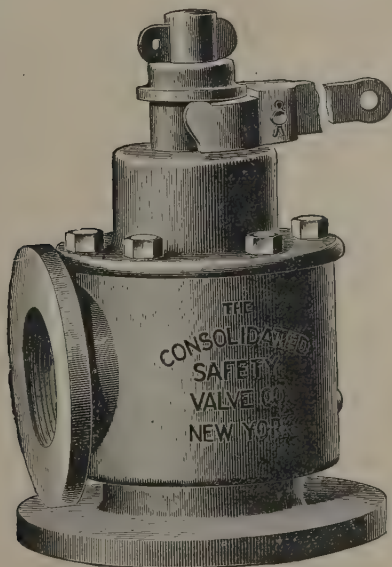


Fig. 55. Pop Valve.

Most pop valves for stationary boilers are provided with a short lever, and if at any time the valve does not pop when the steam gauge shows the pressure to

be high enough, it can generally be started by a light blow on the lever with a hammer.

The ratio of safety valve area to that of grate surface is, for the old style lever and weight valve, 1 sq. in. of valve area for each 2 sq. ft. of grate surface, and for pop valves 1 sq. in. of valve area for each 3 sq. ft. of grate surface.

Each boiler in a battery should have its own safety valve, and, in fact, be entirely independent of its mates as regards safety appliances.

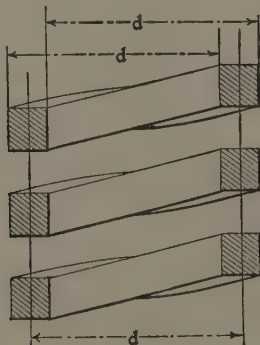


Fig. 56.

One example of safety valve computation will be given. Suppose the grate surface of a boiler is $5 \times 6 = 30$ sq. ft., what should be the diameter of the lever safety valve? The required area of the valve is $30 \div 2 = 15$ sq. in. Then $15 \div .7854 = 19$, which is the square of the diameter of the valve. Extracting the square root of 19 gives 4.35 in. diameter of valve. In actual practice one 5 in. or two 3 in. lever safety valves would be required.

If a pop valve is to be used the required area is $30 \div 3 = 10$ sq. in. Then $10 \div .7854 = 12.73 = \text{square of diameter}$

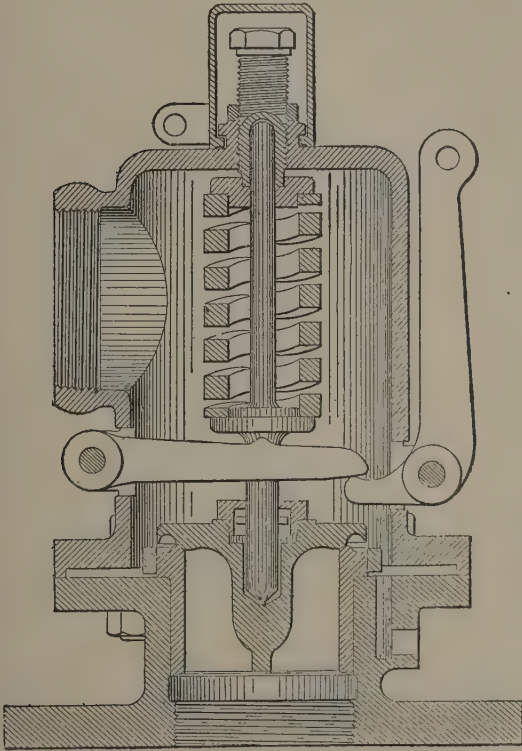


Fig. 57.

of valve. Extract the square root of 12.73 and the result is 3.6 in. = diameter of valve. In practice a 4 in. valve would be required.

Regarding the pressure at which a spring-loaded or pop safety valve will blow off, it is first necessary to ascertain by experiment the force required to compress the spring. The pressure at which a spring will yield depends not only upon the shape and size of the material of which it is made, the diameter, number and pitch of the coils, all of which are measurable and determinable, but upon the nature and condition of the material itself.

For instance, a spring of brass will compress with less pressure than one of steel, similar in every other respect, and there is such a wide difference in steels that there will be a great deal of difference in the action of steel springs according to the kind of metal, degree of temper, etc.

The following rule may be used in calculations of this character:

RULE.

Multiply the compression in inches by the fourth power of the thickness of the steel in sixteenths of an inch, and by 22 for round or 30 for square steel. Product I.

Multiply the cube of the diameter of the spring, measured from center to center of the coil (as on the line d, in Fig. 56) in inches, by the number of free coils in the spring, and by the area of the valve in square inches. Product II.

Divide Product I by Product II and the quotient will be the pressure per square inch at which the valve will blow off.

With a dead weight or a lever-loaded valve the force required to lift it remains the same, no matter how high the valve lifts. The weights weigh no more if they are

raised an inch or two, and the leverage does not change, but with the spring-loaded valve, the more the valve lifts

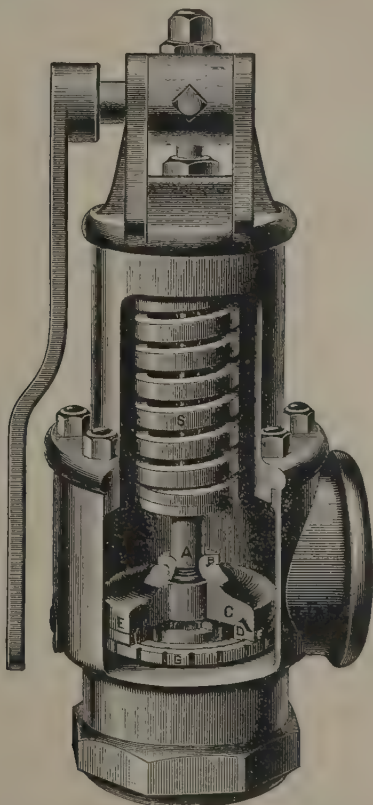


Fig. 58. Inside View of a Pop Safety Valve.

the more the spring is compressed, and the more force is required to compress or hold it. It follows then that

if an ordinary valve were loaded with a spring it would simply crack open and commence to sizzle when the pressure equaled the force at which the spring was set, and that if this were not enough to relieve the boiler the pressure would have to increase, opening the valve more and more until the steam blew off as fast as it was made.

But the ideal valve should stay on its seat until the pressure reaches the desired limit, then open wide and discharge the excess. This result is accomplished by the construction shown in Fig. 57. With the first opening of the valve the steam passes into the little "huddling chamber" made by the cavity near the overhanging edge of the valve, and a similar cavity surrounding the seat. The pressure which accumulates here, acting on the additional area of the valve, raises it sharply with the "pop" which gives the valve its name, and it is sustained by the impact and reaction of the issuing steam until the pressure has subsided sufficiently to allow the spring to overcome these actions.

A short space will be devoted to the consideration of the lever safety valve also, as it may be of interest to some students.

The U. S. marine rule for lever valves is here repeated: "Lever safety valves to be attached to marine boilers shall have an area of not less than one square inch to every two square feet of grate surface in the boiler, and the seats of all such safety valves shall have an angle of inclination of 45° to the center line of their axis."

In order to arrive at accurate results in lever safety valve calculations it is necessary to know first the number of pounds pressure exerted upon the stem of the valve by the lever itself, irrespective of the weight, also

the weight of the valve and stem, as all these weights together with the weight of the ball suspended upon the lever tend to hold the valve down against the pressure of the steam. The effective weight of the lever can be ascertained by leaving it in its position attached to the fulcrum and connecting a spring balance scale to it at a point where it rests on the valve stem. The weight of the valve and stem can also be found by means of the scale. When the above weights are known, together with the weight at the end of the lever and its distance from the fulcrum, also the area of the valve and its distance from the fulcrum, the pressure at which the valve will blow can be found by the following rules:

Rule 1. Multiply the weight by its distance from the fulcrum. Multiply the weight of the valve and lever by the distance of the stem from the fulcrum and add this to the former product. Divide the sum of the two products by the product of the area of the valve multiplied by the distance of its stem from the fulcrum. The result will be pressure in pounds per square inch required to lift the valve.

Example. Diameter of valve, 3 in.

Distance of stem from fulcrum, 3 in.

Effective weight of lever, valve and stem, 20 lbs.

Weight of ball, 50 lbs.

Distance of ball from fulcrum, 30 in.

Required pressure at which the valve will blow off.

$$50 \times 30 + 20 \times 3 = 1560.$$

$$\text{Area of valve, } 7.0686 \times 3 = 21.2058.$$

$$1560 \div 21.2058 = 73.57 \text{ lbs. pressure.}$$

When the pressure at which it is desired the valve should blow off is known, together with the weights of

all the parts, the proper distance from the fulcrum at which to place the weight is ascertained by Rule 2.

Rule 2. Multiply the area of the valve by the pressure and from the product subtract the effective weight of the valve and lever. Multiply the remainder by the distance of the stem from the fulcrum and divide by the weight of the ball. The quotient will be the required distance.

Example. Area of valve, 7.07 sq. in.

To blow off at 75 lbs.

Effective weight of lever and valve, 20 lbs.

Weight of ball, 50 lbs.

Distance of valve stem from fulcrum, 3 in.

$7.07 \times 75 - 20 = 510.25.$

$510.25 \times 3 \div 50 = 30.6$ in., distance from fulcrum at which to place the ball.

When the pressure is known, together with the distance of the weight from the fulcrum, the weight of the ball is obtained by Rule 3.

Rule 3. Multiply the area of the valve by the pressure and from the product subtract the effective weight of the lever and valve. Multiply the remainder by the distance of the stem from the fulcrum and divide by the distance of the ball from the fulcrum. The quotient will be the required weight.

Example. Area of valve 7.07 sq. in.

Pressure in pounds per square inch..... 80 lbs.

Effective weight of lever and valve..... 20 lbs.

Distance of stem from fulcrum..... 3 in.

Distance of weight from fulcrum..... 30 in.

$7.07 \times 80 - 20 = 545.6.$

$545.6 \times 3 \div 30 = 54.56$ lbs., weight of ball.

Safety valves, especially those of the lever type, are liable to become corroded and stick to their seats if allowed to go any great length of time without blowing. Therefore it is good practice to raise the steam pressure to the blowing off point at least two or three times a week, or oftener, for the purpose of testing the valve. If it opens and releases the steam at the proper point all is well, but if it does not, it should be looked after forthwith. Generally the mere raising of the lever by hand, or a few taps with a hammer if it be a pop valve, will free it and cause it to work all right again; but if this treatment has to be resorted to very often the valve should be taken down and overhauled. In too many steam plants not enough importance is attached to the safety valve. The fact is, it is one of the most useful and important adjuncts of a boiler, and if neglected serious results are sure to follow.

STEAM GAUGES.

There are different types of steam gauges in use, but the one most commonly used, and which no doubt is the most reliable, is known as the Bourdon spring gauge. This gauge consists of a thin, curved, flattened metallic tube, closed at both ends and connected to the steam space of the boiler by a small pipe, bent at some portion of its length into a curve or circle that becomes filled with water of condensation, and thus prevents the hot live steam from coming directly in contact with the spring, while at the time the full pressure of steam in the boiler acts upon the spring, tending to straighten it. The end or ends of the spring being free to move, and connected by suitable geared rack and pinion with the pointer of the gauge, this hand or pointer is caused to move across

the dial, thus indicating the pressure of steam per square inch in the boiler. When there is no pressure in the boiler the hand should point to 0.

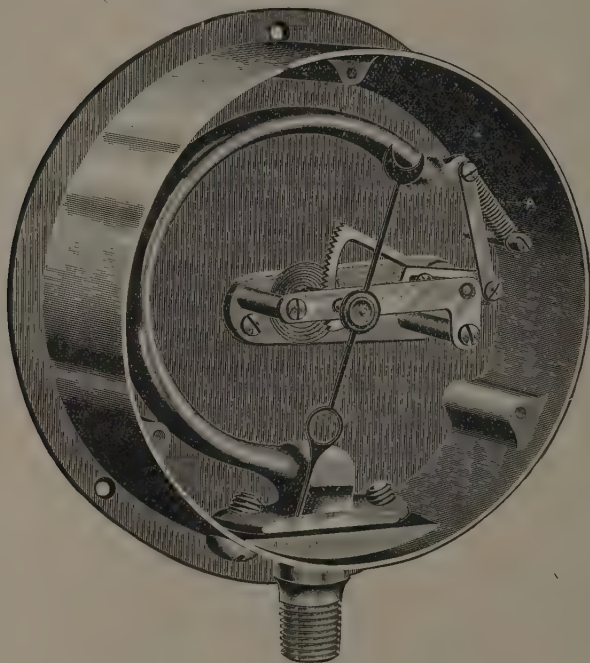


Fig. 58a. Auxiliary Spring Pressure Gauge, Sectional View.

Steam gauges should be tested frequently by comparing them with a test gauge that has been tested against a column of mercury.

There should always be a trap or siphon in the gauge pipe in order to retain the water of condensation, so as

to prevent the hot steam from coming in contact with the spring.

If at any time the water is drained from the siphon, care should be exercised in turning on the steam again, by allowing it to flow in very slowly at first until the siphon is again filled with water.



Fig. 58b. Auxiliary Spring Pressure Gauge.

The steam gauge and the safety valve should be compared frequently by raising the steam pressure high enough to cause the valve to open at the point for which it is set to blow.

BURSTING PRESSURE AND SAFE WORKING PRESSURE OF
BOILERS.

Bursting Pressure. The query might arise, why should the longitudinal or side seams require to be stronger than the girth or round about seams? The answer is, that the force tending to rupture the boiler along the line of the longitudinal seams is proportional to the diameter divided by two, while the stress tending to pull it apart endwise is only one-half that, or proportional to the diameter divided by four.

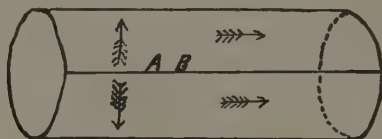


Fig. 59.

To illustrate, let Fig. 59 represent the shell of the boiler heretofore referred to, ignoring for the time being the tubes and braces, and consider the boiler simply as a hollow cylinder. Now the total force tending to rupture the boiler along the line of the girth seams or in the direction of the horizontal arrows = area of one head in square inches $\times$ pressure in pounds per square inch. It is true that the pressure is exerted against both heads, but the area of one head can only be considered for the reason that the two stresses are exerted against each other just as in the case of two horses pulling against each other, or in opposite directions on the same chain. The stress on the chain will be what each horse (not both) pulls. To further illustrate, suppose one of the

horses to be replaced by a permanent post or wall and let one end of the chain be attached thereto. One head or one side of the boiler pulls against the other, and the stress on the seams is the force with which each (not both) pulls. Referring again to Fig. 59, area of one head $= 60^2 \times .7854 = 2827.4$ sq. in. Suppose there is a pressure of 10 lbs. per square inch in the boiler. Then total stress on the girth seams $= 2827.4 \times 10 = 28,274$ lbs. Opposed to this pull is the entire circumference of the boiler, which is $60 \times 3.1416 = 188.5$ in. Therefore, dividing total pressure (28,274 lbs.) by the circumference in inches (188.5) will give 150 lbs. as the stress on each inch of the girth seams. While the stress on each inch of the longitudinal seams or along the line A B, Fig. 59, and which is exerted in the direction of the vertical arrows, is pressure (10 lbs.) $\times$ one-half the diameter (30 in.) $= 300$ lbs. One-half the diameter is used because the pressure in any direction is effective only on the surface at right angles to that direction.

The formula for finding the bursting pressure of a boiler may be expressed as follows:

$$B = \frac{T.S. \times T}{R} \text{ in which } B = \text{bursting pressure.}$$

T.S. = tensile strength.

T = thickness of sheet.

R = radius or one-half the diam.

Example. T. S. = 55,000 lbs. per square inch.

T = $\frac{3}{8}$ in. (expressed decimally = .375 in.).

R = 30 in.

Then $55,000 \times .375 \div 30 = 687.5$ lbs. per square inch, which is the pressure at which rupture would take place

provided there were no seams in the boiler and the original strength of the sheet was retained, but, as has been seen, a certain percentage of strength is lost through punching or drilling the necessary rivet holes, and this must be taken into account.

The formula now becomes, for double riveting, $B = \frac{T.S. \times T \times .70}{R}$, in which the letters preserve the same value as in the original formula, but the result is reduced by multiplying by the decimal .70, which represents the percentage of strength retained by double-riveted seams. Consequently B will now =

$$\frac{55,000 \times .375 \times .70}{30} = 481 \text{ lbs.}$$

In case the seams are all single riveted .56 must be substituted for .70, and with triple-riveted butt joints .88 can safely be used.

Safe Working Pressure. In order to ascertain the safe working pressure of a boiler it is necessary first to calculate the bursting pressure and divide this by another factor called the factor of safety. The one most commonly used for boilers is 5, or in other words the safe working pressure = one-fifth the bursting pressure. In the case of the boiler under consideration, the safe pressure would be $481 \div 5 = 96$ lbs., at which point the safety valve should blow off.

Fusible Plugs. A fusible plug should be inserted in that part of the heating surface of a boiler which is first liable to be overheated from lack of water.

In a horizontal tubular or return flue boiler the proper location for the fusible plug is in the back head about $1\frac{1}{2}$ or 2 in. above the top row of tubes. In fire-box boilers

the plug can be put into the crown sheet directly over the fire. These plugs should be made of brass with hexagon heads and standard pipe threads, in sizes $\frac{1}{2}$, $\frac{3}{4}$, 1 in., or even larger if desired. A hole drilled axially through the center and counter sunk in the end that enters the boiler is filled with an alloy of such composition that it will melt and run out at the temperature of the dry steam at the pressure carried in the boiler. Thus, if the water should get below the plug the dry steam, coming in contact with the fusible alloy, melts it and, escaping through the hole in the plug, gives the alarm, and in case of fire-box or internally fired boilers the steam will generally extinguish the fire also. The hole is counter sunk on the inner end of the plug so as to retain the fusible metal against the boiler pressure. These plugs should be looked after each time the boilers are washed out, and all dirt and scale should be cleaned off in order that the fusible metal may be exposed to the heat.

Another type of fusible plug consists of a small brass cylinder into one end of which is screwed a plug filled with a metal which will fuse at the temperature of dry steam at the pressure which is to be carried in the boiler. The other end of the cylinder is reduced and fitted with a small stop valve and threaded to screw into a brass bushing inserted into the top of the boiler shell. This bushing also receives at its lower end a piece of $\frac{1}{2}$ or $\frac{3}{4}$ in. pipe which extends downwards to within 2 in. of the top row of tubes, or the crown sheet, if the boiler is internally fired. The principle of the device is that in case the water falls below the lower end of the pipe, steam will enter, fuse the metal in the plug, and be free to blow and give warning of danger. Some of these appliances are fitted with whistles which are sounded in case the

steam gets access to them. But even with such devices no engineer can afford to relax his own vigilance, and depend entirely upon the safety appliances to prevent accidents from low water.

Feed Pipes. Authorities differ in regard to the proper location of the inlet for the feed pipe, but upon one point all are agreed, namely, that the feed water, which is always at a lower temperature than the water in the boiler, should not be allowed to come directly in contact with the hot boiler sheets until its temperature has been raised to within a few degrees of the temperature of the water in the boiler. Certainly one of the most fruitful sources of leaks in the seams, and around the rivets is the practice of introducing the feed water into the bottom either at the back or front ends of boilers, as is too often the case. The cool water coming directly in contact with the hot sheets causes alternate contraction and expansion, and results in leaks, and very often in small cracks in the sheet, the cracks extending radially from the rivet holes. It would appear that the proper method is to connect the feed pipe either into the front head just above the tubes, or into the top of the shell. The nipple entering the boiler should have a long thread cut on the end which screws into the sheet, and to this end inside the boiler there should be connected another pipe which shall extend horizontally at least two-thirds of the length of the boiler, resting on top of the tubes, and then discharge. Or, what is still better, allow the internal pipe to extend from the entering nipple at the front end to within a few inches of the back head, then at right angles across the top of the tubes to the other side, and from there discharge downward. By this method the feed water is heated to nearly, if not quite, the temperature of the

water in the boiler before it is discharged. One of the objections to this system is the liability of the pipe inside the boiler to become filled with scale and finally plugged entirely. In such cases the only remedy is to replace it with new pipe. But the great advantage of having the water thoroughly heated before being discharged into the boiler will much more than compensate for the extra expense of piping, and the general idea of introducing the feed water at the top, instead of at the bottom of the boiler is therefore recommended as being the best.

The diameter of feed pipes ranges from 1 in. for small sized boilers, up to 1½ and 2 in. for boilers of 54 to 72 in. in diameter. It is not good policy to have the feed pipe larger than necessary for the capacity of the boiler; because it then acts as a sort of cooling reservoir for the feed water, and may cause considerable loss of heat.

For batteries of two or more boilers it is necessary to run a main feed header, with branch pipes leading to each boiler. The header should be large enough to supply all the boilers at the same time, should it ever become necessary to do so. The header can be run along the front of the boilers just above the fire doors, with the branch pipes running up on either side, clear of the flue doors and entering the front connection, or smoke arch, and the boiler head at a point two inches above the tubes. There should always be a valve in each branch pipe between the check valve and the header for the purpose of regulating the supply of water to each boiler, and also for shutting off the pump pressure in case of needed repairs to the check valve. Another valve should be placed between the check valve and the boiler. By this arrangement it is always possible to get at the check valve when it is out of order.

Blow Off Pipes. Blow off pipes should always be connected with the lowest part of the water space of a boiler. If there is a mud-drum, then of course the blow off should be connected with it; but if there is no mud-drum, the blow off should connect with the bottom of the shell, near the back head, extend downwards to the floor of the combustion chamber, and thence horizontally cut through the back wall, where the blow off cock can be located.

The best blow off cocks are the asbestos packed, iron-body plug cocks, which are durable and safe. A globe valve should never be used in a blow off pipe, because the scale and dirt will lodge in it and prevent its being closed tightly. A straight way or gate valve is not so bad, but an asbestos packed plug cock is undoubtedly the best and safest.

In order to protect the blow off pipe from the intense heat, a shield consisting of a piece of larger pipe can be slipped over the vertical part before it is connected.

Blow off cocks should be opened for a few seconds once or twice a day, to allow the scale and mud to be blown out. If neglected too long they are liable to become filled with scale and burn out. A plan which is said to give good results is to connect a tee in the horizontal part of the pipe, and from this tee run a 1 in. pipe to a point in the back head at the water level. It is claimed that this will cause a circulation of water in the pipe and prevent the formation of scale.

A surface blow off is a great advantage, especially if the water is muddy or liable to foam. By having the surface blow off connected on a level with the water line a large amount of mud, and other matter which is kept on the surface by the constant ebullition can be blown out.

A combination surface blow off, bottom blow off and circulating system can be arranged by a connection such as illustrated in Fig. 60. By closing cock A and opening cocks B and C the bottom blow off is put in operation; by closing B and opening A and C the surface blow off is started, and by closing C and leaving A and B

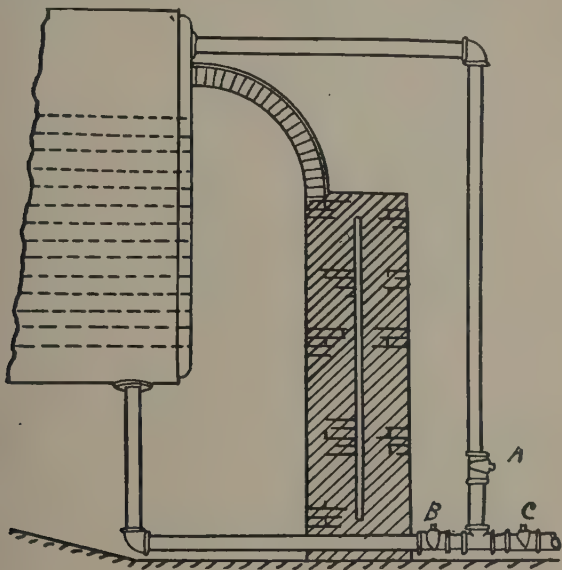


Fig. 60.

open the device will act as a circulating system. The pipe should be of the same size throughout. Blow off pipes should be of ample size, never less than $1\frac{1}{4}$ in., and from that to $2\frac{1}{2}$ in., depending upon the size of the boiler.

Feed Pumps. The belt driven power pump is the most economical boiler feeder, but is not the most convenient nor the safest. When the engine stops, the pump stops also, and sometimes it happens that the belt gives way and the pump stops at just the time when the boiler is being worked the hardest.

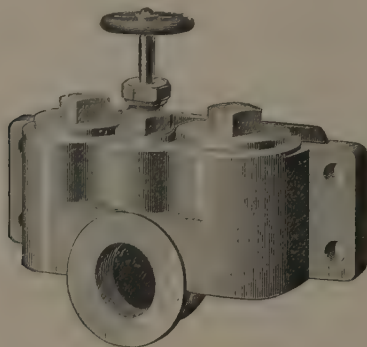


Fig. 61. Differential Valve, Davis Pump.

The modern double acting steam pump, of which there are many different makes to choose from, is without doubt the most reliable boiler feeding appliance, and the one best adapted to all circumstances and conditions, although it is not economical in the use of steam, since the principle of expansion cannot be carried out with the pump as with the engine.

In selecting a feed pump care should be exercised to see that it is of the proper size and capacity to supply the maximum quantity of water that the boiler can evaporate. This may be ascertained by taking into consideration the amount of heating surface and the required

consumption of coal per square foot of grate surface per hour. First, take the coal consumption. Assume the boiler to have 30 sq. ft. of grate surface, and that it is desired to burn 15 lbs. of coal per square foot of grate per hour, which is a good average with the ordinary hand fired furnace using bituminous coal. Suppose the boiler is capable of evaporating 8 lbs. of water per pound of coal consumed. Then $30 \times 15 \times 8 = 3,600$ lbs. of

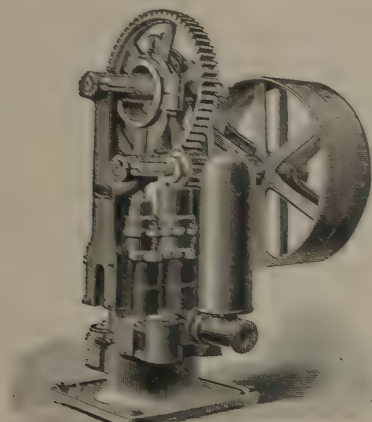


Fig. 62. Davis Belt Driven Feed Pump.

water evaporated per hour. Dividing 3,600 by 62.4 (the weight of a cubic foot of water in pounds) gives 57.6 cu. ft. per hour, which, divided by 60, gives 0.96 cu. ft. per minute. This multiplied by 1,728 (number of cubic inches in a cubic foot) gives 1,659 cu. in. per minute which the pump is required to supply. Suppose the pump is to make forty strokes per minute, and the length of stroke is five inches. Then $1,659 \div 40 = 41.47$ cu. in.

per stroke, which, divided by 5 (length of stroke in inches) gives 8.294 sq. in. as the required area of water piston. $8.294 \div .7854 = 10.56$, which is the square of the corresponding diameter, and the square root of $10.56 = 3.25$. So, theoretically, the size of the water end of the pump would be $3\frac{1}{4}$ in. in diameter by 5 in. stroke; but as it is always safer to have a reserve of pumping capacity, the proper size of the pump would be $3\frac{1}{2}$ in. in diameter, by 5 in. stroke, with a steam cylinder of 6 or 7 in. in diameter.

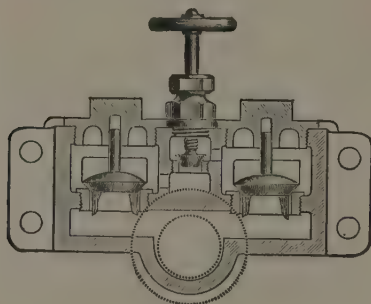


Fig. 63. Sectional View of Differential Valve.

There is another rule for ascertaining the size of the feed pump, viz., by taking the number of square feet of heating surface in the boiler and allow a pump capacity of 1 cu. ft. per hour for each 15 sq. ft. of heating surface. Thus, let the total heating surface of the boiler be 786 sq. ft. Dividing this by 15 gives 52.4 as the number of cubic feet of water required per hour, from which the pump dimensions may be found in the same way as in the preceding case.

In figuring on the capacity of a feed pump for a battery of two or more boilers, the total quantity of water required by all the boilers must be taken into consideration. All boiler-rooms should be supplied with at least two feed pumps, so that if one breaks down there may always be another one available.



Fig. 64. Worthington Duplex Boiler Feed Pump.

Hard rubber valves are, all things considered, the best for a boiler feed pump, as they are not affected by hot water and do not hammer the seats like metallic composition valves do. Every boiler feed pump should be fitted with a good sight-feed lubricator for cylinder oil. The steam valve mechanism of a steam pump is very sensitive and delicate, and requires good lubrication in order to do good work. In too many cases feed pumps are fitted with an old style cylinder oil cup and there is generally more oil on the outside of the valve chest than there is inside, while the valve is bulldozed into working by frequent blows from a convenient slab.

The steam valves of all steam pumps are adjusted before they are sent out from the factory, and most of them are arranged so that the stroke may be shortened or lengthened as the engineer desires. It is best, as a

rule, to allow a pump to make as long a stroke as it will without striking the heads, because then the parts are worn evenly.

Sometimes an engineer is called upon to set the valves of a duplex pump which have become disarranged. In such a case he should proceed as follows: Place both pistons exactly at mid-stroke. This may be done in two ways. First, by dropping a plummet line alongside the levers connecting the rock shafts with the spools on the

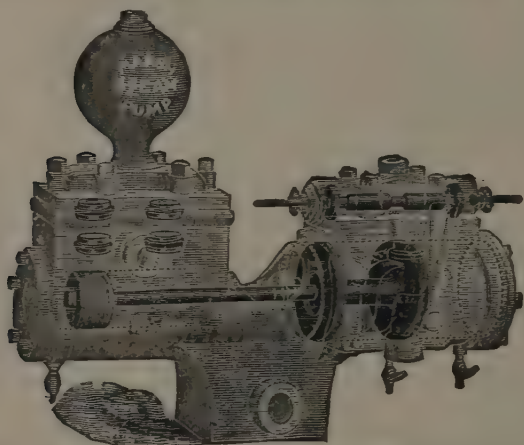


Fig. 65. Phantom View of Marsh Independent Steam Pump.

piston rods. Then bring the rods to the position where the centers of the spools will be in a vertical line with the centers of the rock shafts.

The second method is to move the piston to the extreme end of the stroke until it comes in contact with the cylinder head. Then mark the rod at the face of the stuffing box gland. Next move the piston to the

other end of the stroke and mark the rod at the opposite gland. Now make a mark on the rod exactly half way between the two outside marks and move the piston back until the middle mark is at the face of the gland and the piston will be at mid-stroke. Having placed both pistons at mid-stroke, remove the valve chest covers and adjust the valves in their central position, viz., so

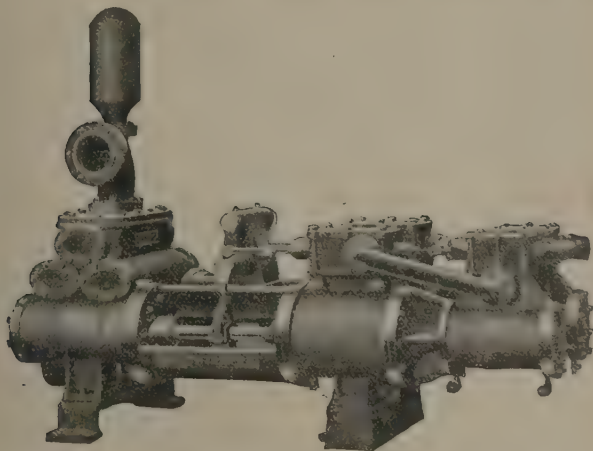


Fig. 66. THE WORTHINGTON COMPOUND STEAM PUMP.
Piston Pattern, for General Service—For 150 Pounds Water Pressure.

that they cover the steam ports. The valve rod being in position and connected to the rocker arms by means of the short link, the nut or nuts securing the valve to the rod should be so adjusted as to be equidistant from the lugs on the valve, say $3/32$ or $1/8$ of an inch, according to the amount of lost motion desired, which latter factor governs the length of stroke in some makes of duplex pumps, while in others it is controlled by tappets

on the valve rod outside of the valve chest. Care should be taken while making these adjustments that the valve be retained exactly in its central position.

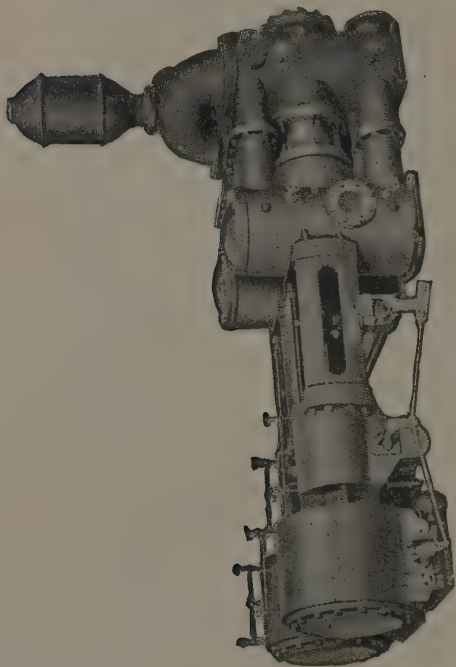


Fig. 67. THE WORTHINGTON PACKED-PLUNGER PUMP.
Scranton Pattern—For 250 Pounds Water Pressure.

Having set the valves correctly, move one of the pistons far enough from mid-stroke to get a small opening of the steam port on the opposite side, then replace the valve chest covers and the pump will be ready to run. As these valves are generally made without any outside

lap, a slight movement of one of the pistons in either direction from its central position will suffice to uncover one of the ports on the other cylinder sufficiently to start the pump.

Sometimes duplex pumps "work lame," that is, one piston will make a quick full stroke, while the other piston will move very slowly and just far enough to work the steam valve of the opposite side. In the majority of cases this irregular action is due to unequal friction in the packing of the rods, or the packing rings on one of the pistons may be worn out.

If one side of a duplex pump becomes disabled from any cause, as breaking of piston rod in the water cylinder, for instance, which is liable to happen, the pump may still be operated in the following manner until duplicate parts to replace the broken ones have been secured: Loosen the nuts or tappets on the valve stem of the broken side and place them far enough apart, so that the steam valve will be moved through only a small portion of its stroke, thereby admitting only steam enough to move the empty steam piston and rod, and thus work the steam valve of the remaining side. The packing on the broken rod should be screwed up tight, so as to create as much friction as possible; there being no resistance in the water end. In this way the pump may be operated for several days or weeks and thus prevent a shut down.

LARGE BOILER FEED PUMPS.

The plan of using one large pump for feeding the boilers has recently been tried in several large power stations. For instance, in the Ashley Street Power House of the Union Electric Light and Power Company

of St. Louis, there was recently installed one large Prescott steam pump to take the place of four smaller ones which had been in use for feeding the boilers.

This pump is of the duplex compound condensing type, having high-pressure steam cylinders 18 inches in diameter, low-pressure steam cylinders 34 inches in diameter and water plungers 17 inches in diameter, with 24-inch stroke in each instance. The normal capacity of the pump is 1,800 gallons per minute, or 900,000 pounds of water per hour, which is equal to 45,000 kilowatts at 20 pounds per kilowatt-hour. The present capacity of the Ashley street power station is 36,000 kilowatts.

The novel departure of installing a boiler feed-pump in one large unit for this work will arouse the interest of engineers, because it is customary to install several small units for boiler-feed purposes in power plants of this size. But as a matter of fact, this large pump is the consummation of a process of evolution in this plant, which was at first equipped with several small feed-pumps.

To produce the most economical results, the low-pressure cylinders and their heads are steam-jacketed; the steam valves are of the rotative type, there being one steam valve for the high-pressure cylinders, and two each for the low-pressure cylinders. The arrangement of two steam valves in the low-pressure steam cylinders reduces the clearance to the lowest possible amount. The water end is of the pot-form type, having four suction and discharge valves, $5\frac{3}{4}$ inches in diameter, in each quarter of the suction and discharge. The water plungers are of cast bronze of the outside end-packed type.

The boiler pressure carried is 175 pounds per square

inch. The pump operates against a pressure of 225 pounds per square inch in the feed-water pipe system to the boilers. This feed-water system is provided with a relief valve, and the pump is controlled by a pressure governor. The boilers are provided with thermostat valves, which allow the water to flow into the boilers and maintain the proper level. All of the automatic apparatus may be controlled by hand, when desirable, for all the boilers, or for each boiler separately. Such occasions are very rare, however, as the pump responds readily to changes in station load, and is under perfect control of the automatic devices for controlling and delivering the proper amount of feed-water to each boiler.

The feed-pipe system, in effect, is simply a system of water mains, in which the number of valves and fittings liable to leak are reduced to a minimum.

Figures 66 and 67 show views of the Worthington boiler feed pump, adaptable to large steam plants. In the arrangement of steam cylinders shown in figure 66, the steam is used expansively, thereby effecting a great saving.

Figure 68 illustrates a type of pump which is rapidly coming into favor for feeding boilers against high pressure. The steam pressures of from 150 to 200 pounds, which are in common use, require a more substantial construction of feed-pump than the lower pressures of a few years ago. These machines are designed for high pressure and for handling either hot or cold water. The steam ends are made extra heavy and provided with extra strong bolting for all joints, making them suitable for constant operation under steam pressures up to 200 pounds per square inch. The water ends are of the piston pattern, pot-valve type, are of ample strength

for working pressure up to 200 pounds and will easily stand test pressure of 300 pounds per square inch. The piston-pattern construction requires less room than the

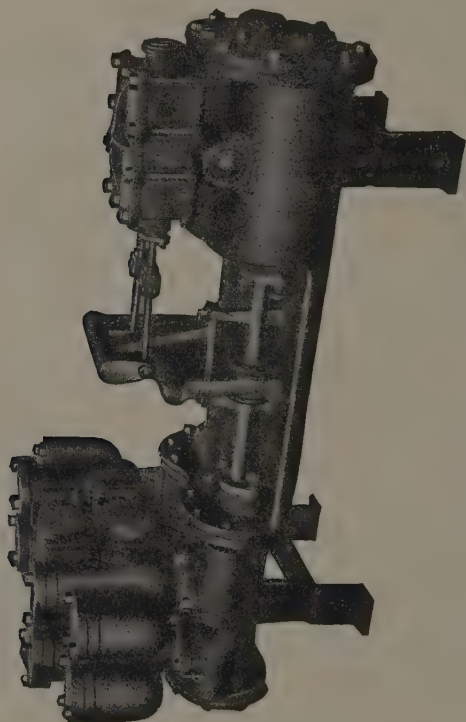


Fig. 68. Deane Piston-Pattern Pot-Valve-Type Boiler Feed-Pump.

outside-packed machine of either the center-packed or end-packed type, and, furthermore, does away with the large outside stuffing-boxes and excessive amounts of

drip. The pistons are fibrous-packed and are readily accessible on removal of the cylinder-heads.

The valves are in special valve chambers or pots, located above the cylinders. This arrangement provides for constant submergence of the pistons, and the reliability of action of the machine consequent to such submergence. Each pot contains one suction and one discharge valve, each valve having an individual cover easily removable for inspection. Valves are of composition of the wing-guided type, with bevel seats, and are of ample area for the requirements of the service. A manifold connects the suction openings of the various pots to a common suction inlet and another manifold provides a common discharge outlet.

While designed primarily for boiler-feed purposes, these pumps are very desirable machines for general service against pressures not exceeding 200 pounds per square inch.

HYDRAULICS FOR ENGINEERS.

Among the many difficult problems that are continually coming up for engineers to solve, there is none more perplexing than the correct calculation of the quantity of water which will be discharged in a given time from pipes of various sizes, and under the many different heads or pressures. Problems in hydraulics, as given by the majority of writers on engineering, are usually in elaborate algebraic equations, which, to the ordinary working engineer, are very perplexing, at least the author has found them to be so in his experience. Therefore with a view of assisting his brother engineers in the solution of problems along this line which they may be called upon to solve, the author has spent considerable time and labor in searching for and compiling a few rules and examples for hydraulic calculations in plain arithmetic, which he hopes may be of benefit.

First, to find velocity of flow in the pump, or in other words, piston speed.

Rule. Multiply number of strokes per minute by length of stroke in feet, or fractions thereof.

Second, the velocity of flow in the discharge pipe is in inverse ratio to the squares of the diameters of the pipe and the water cylinder of pump.

Thus, a pump cylinder is 6 in. in diameter, and the piston speed is 100 ft. per minute, the discharge pipe being 3 in. in diameter. What is the velocity of flow in the pipe?

Example. $\frac{6 \times 6}{3 \times 3} = 4$ In this case the velocity in the

pipe is four times that in the pump, and $100 \div 4 = 400$ ft. per minute, velocity for water in the discharge pipe.

Third, to find velocity in feet per minute necessary to discharge a given quantity of water in a given time.

Rule. Multiply the number of cubic feet to be discharged by 144 and divide by area of pipe in inches.

Fourth, to find area of pipe when the volume and velocity of water to be discharged are known.

Rule. Multiply volume in cubic feet by 144 and divide by the velocity in feet per minute.

Fifth, one of the first requisites in making correct calculations of the quantity of water discharged from any sized pipe is to obtain the velocity of flow per second. There are several rules for doing this, among which the following appear to be the plainest and most simple:

Rule 1. Multiply the square root of the head in inches by the constant 27.8. For instance, assume the head to be 100 ft. = 1,200 in. The square root of 1,200 is 35 nearly, then $35 \div 27.8 = 973$ in. = 81 ft. per second velocity.

Rule 2. Multiply the square root of the head in feet by the constant 8, as follows: The square of 100 = 10 and $10 \div 8 = 80$ ft. velocity per second.

Rule 3. Multiply twice the acceleration of gravity by the head in feet and extract the square root of product. The acceleration of gravity may be considered the constant number 32, neglecting decimals, $32 \div 2 \div 100 = 6,400$. Square root of 6,400 = 80 ft. per second.

In many instances it is more convenient to use the pressure in pounds per square inch as shown by gauge instead of the height or head, and we can then apply Rule 4.

Rule 4. Multiply the square root of the pressure in pounds per square inch by the constant number 12.16 as follows: Pressure due to 100 ft. head = 44 lbs., nearly. Square root of 44 = 6.6, which multiplied by 12.16 = 80.2 ft. velocity per second.

Having ascertained the velocity of flow, we may now proceed to calculate the weight of water in pounds per second discharged from any size of pipe, neglecting for the time being the loss in pressure caused by friction from elbows and bends in the pipe and also the peculiar shape assumed by a stream of water flowing through pipes or conduits when there is no resistance except the pressure of the atmosphere and friction caused by long distance transmission.

We will take for our calculation a four-inch pipe from which the water has a free flow under a head of 100 ft., which gives a velocity of 80 ft. per second.

Rule 5. Divide the velocity in feet per second by the constant 2.3, and multiply the quotient by the area of discharge pipe in square inches. $80 \div 2.3 = 34.7$. Now the area of a four-inch pipe is 12.57 sq. in., and $34.7 \times 12.57 = 436$ lbs. discharged per second.

In order to get the matter clearly before us, let us assume that we have a section of four-inch pipe just 80 ft. in length and that it lies in a horizontal position and is filled solidly full of water. It will contain area, 12.57 sq. in. $\times$ length, 960 in. = 12,067.2 cu. in. of water, and as one pound of water occupies a space of 27.7 cu. in., we therefore have $12,067.2 \div 27.7 = 436$ lbs. of water, and at a velocity of 80 ft. per second our pipe will be emptied and refilled continuously each second. We have also Rule 6 to find the number of

cubic feet discharged per minute when the velocity per minute is known.

Rule 6. Multiply the area of pipe in square inches by the velocity in feet per minute and divide by the constant 144.

Example. Area of 4 in. pipe = 12.57 sq. in. Velocity of flow = 80 ft. per second = 4,800 ft. per minute.

Then $\frac{12.57 \times 4,800}{144} = 419$ cu. ft. per minute = 6.99 cu.

ft. per second, which multiplied by 62.3 lbs. (weight of 1 cu. ft.) = 435.4 lbs. per second.

As stated before, no allowance is made by the above rules for friction or other retarding influences, but for ordinary purposes in connection with a steam plant a deduction of 25 per cent is probably sufficient. Of course if the water is being discharged into an elevated tank or against direct pressure of any kind, the resistance in pounds per square inch or the height in feet must be deducted from the impelling pressure or head. Let us assume, for instance, that our 4 in. pipe is discharging water into a tank at an elevation of 75 ft. above the level of the pump, and that to reach the tank requires 100 ft. of pipe with two 90° ells and one straight-way valve. We wish to discharge 500 gals. per minute into the tank, and will therefore require a velocity of about 13 ft. per second, which will necessitate a pressure of a little more than 1 lb. per square inch to be maintained at the pump over and above all resistance. Now the resistance to be overcome in this case will be:

Pressure per square inch due to 75 ft. head. . . .	32.5 lbs.
Friction loss due to length of pipe and velocity. . .	7.43 "
Friction loss due to two 90° ells.	2.16 "
Friction loss due to straight-way valve.	.2 "
Total	42.29 lbs.

Requiring a pressure of say 43 lbs. per square inch, or about the equivalent of 100 ft. head at the pump.

Again, suppose that in place of the elevated tank we have 1,000 ft. of 8 in. horizontal pipe with a 4 in. delivery at the end farthest from the pump, and three branch pipes each 100 ft. long and 4 in. in diameter with one 90° ell one straight-way valve, connected at intervals to the 8 in. main, and it is required to discharge in all 1,000 gals. per minute, or at the rate of 250 gals. per minute for each 4 in. delivery. The friction loss for each 100 ft. in length of 8 in. pipe at a velocity of 13 ft. per second is .94 lbs., and for each 100 ft. of 4 in. pipe it is 1.89 lbs. Likewise the friction loss for each 90° ell is 1.08 lbs., and for a straight way valve .2 lbs., at the above velocity. The total resistance therefore to be overcome is as follows

For 1,000 ft. of 8 in. pipe. . . .	.94 lbs. $\times 10 =$	9.4 lbs.
For 300 ft. of 4 in. pipe.	1.89 lbs. $\times 3 =$	5.67 "
For four 90° ells.	1.08 lbs. $\times 4 =$	4.32 "
For four straight-way valves. . .	.2 lbs. $\times 4 =$	.8 "
Total		20.19 lbs.

Consequently the pressure required at the pump will be about 22 lbs. per square inch, equal to a head of 50 ft.

Connecting a Recently Fired Up Boiler. After a boiler has been washed out, filled with water, and fired

up, the next move is to connect it with the main battery. The steam in the boiler to be connected having been raised to the same pressure as that in the battery, the connecting valve should be opened slightly, just enough to permit a small jet of steam to pass through, which can be heard by placing the ear near the body of the valve. This jet of steam may be passing from the battery into the newly connected boiler or *vice versa*. Whichever way it passes, the valve should not be opened any farther until the flow of steam stops, which will indicate that the pressure has been equalized. It will then be found that the valve will move much easier and it may be gradually opened until it is wide open.

Foaming. Water carried with the steam from the boiler to the engine, even if in small quantities, is very detrimental to the successful operation of the engine, as it washes the oil from the walls of the cylinder, thereby increasing the friction, and unless a plentiful supply of oil is entering the cylinder, cutting of the piston rings will take place. There is also danger of breaking a cylinder head, or of bending the piston rod if the water comes in too large quantities.

There are certain kinds of water which have a natural tendency to foam, especially such as contain considerable organic matter, and the more severe the service to which the boiler is put the more will the water foam, until it is practically impossible to locate the true level of the water in the boilers, and the only recourse the water tender has is to keep his feed pump running at such a speed as will in his judgment supply the water as fast as it goes out of the boilers. It is a dangerous condition, to say the least, and the only remedy for it is either a change to a different kind of

water, or if this is not possible, then an increase in the number of boilers, which would make it possible to supply sufficient steam for the engine without being compelled to fire the boilers so hard.

Priming. By which is meant the carrying over of water in the form of fine spray mingled with the steam, is not so dangerous as foaming, and yet it causes much loss in the efficiency of a boiler or engine. It can be prevented to a large extent by placing a baffle plate in the steam space of the boiler directly under the dome or outlet to the connection with the steam main.

MECHANICAL STOKERS.

The principles governing the operation of mechanical or automatic stokers are in the main correct, viz., that the supply of coal and air is continuous and that provision is made for the regulation of the supply of fuel according to the demand upon the boiler for steam; also that the intermittent opening and closing of the furnace doors, as in hand firing, thereby admitting a large volume of cold air directly into the furnace on top of the fire, is avoided.

Mechanical stokers have within the last twelve years been largely adopted in the United States, especially in sections where bituminous coal is the principal fuel. The disadvantages attending their use are:

First, that the cost of properly installing them is so great that their use is practically prohibited to the small manufacturer.

Second, that in case of a sudden demand upon the boilers for more steam the automatic stoker cannot respond as promptly as in hand firing, although this objection could no doubt be met by skillful handling.

Third, the extra cost for power to operate them, although this is probably offset by the diminished expense for labor required, as compared to hand firing.

There are many different types of mechanical stokers, and automatic furnaces, but they may for convenience be grouped into four general classes. In class one the grate consists of an endless chain of short bars that travel in a horizontal direction over sprocket wheels,

operated either by a small auxiliary engine or by power derived from an overhead line of shafting in front of the boilers.

In class two may be included stokers having grate bars somewhat after the ordinary type as to length and size, but having a continuous motion up and down or forward and back. This motion, though slight, serves to keep the fuel stirred and loosened, thus preventing the firing from becoming sluggish. The grate bars in this class of stokers are either horizontal or inclined at a slight angle, and their constant motion tends to gradually advance the coal from the front to the back end of the furnace.

Class three includes stokers having the grate bars steeply inclined. Slow motion is imparted to the grates, the coal being fed onto the upper end and forced forward as fast as required.

Class four includes an entirely different type of stoker, in that the fresh coal is supplied underneath the grates, and is pushed up through an opening left for the purpose midway of the furnace. The gases, on being distilled, immediately come in contact with the hot bed of coke on top and the result is good combustion. In this type of stoker steam is the active agent used for forcing the coal up into the furnace, either by means of a long, slowly revolving screw, as in the American stoker, or a steam ram, as with the Jones under-feed stoker. A forced draft is employed, and the air is blown into the furnace through tuyeres. When these stokers are intelligently handled they give good results, especially with cheap bituminous coal. The clinker formed on the grate bars or dead plates is easily removed.



Fig. 69.

The coal is supplied to mechanical stokers either by being shoveled by hand into hoppers in front of, and above the grates, or, as is the case in most of the large plants using them, it is elevated by machinery and deposited in chutes, through which it is fed to each boiler by gravity. Stokers of the chain grate variety are usually constructed so that they may be withdrawn from underneath the boiler in case repairs are necessary. The coal, either nut or screenings, is fed into a hopper in

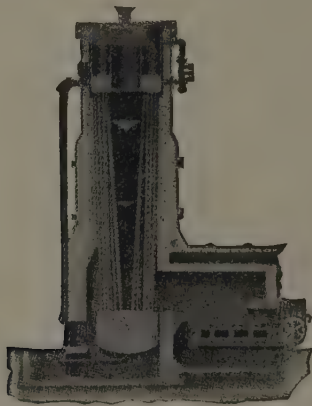


Fig. 70. Cahall Vertical Boiler with Chain Grate Stoker Attached.

front of and above the level of the grate, and is slowly carried along towards the rear end. The ash drops from the grate as it passes over the sprocket wheel at the rear.

Fig. 69 shows a battery of Babcock and Wilcox water tube boilers fitted with chain grate stokers. The buckets for elevating the coal to bins overhead, from whence it is fed by gravity to the stokers, are not

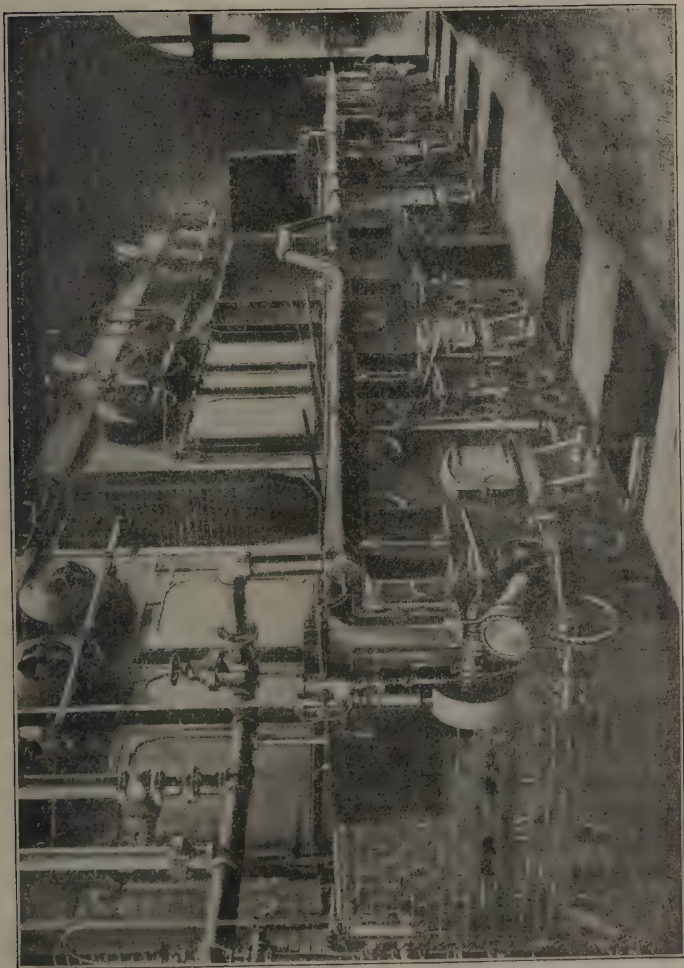


Fig. 71. 1,000 H. P. Cahall Horizontal Boilers.—Cincinnati Street Railway Co.

shown. These buckets or carriers may also be utilized for conveying the ashes from the boiler-room.

Fig. 70 is a sectional view of a vertical Cahall boiler with a Mansfield chain grate stoker, and Fig. 72 shows the same stoker withdrawn from the boiler.

The Coxe mechanical stoker operates upon the same general principles as do those previously described, being of the chain grate type, but it has in addition a series of air chambers just underneath the upper traveling grate. These air chambers are made of sheet iron and are open at the top and provided with dampers for regulating the air pressure for different sections of the grate. The air blast is supplied by a fan. Another feature of this stoker is a water chamber for the bottom section of the grate to travel through on its return.

The Playford stoker has wrought iron T bars extending across the furnace and attached to the traveling chains. These T bars carry the small cast iron sections composing the grate. A screw conveyor is also placed in the ash pit for the purpose of carrying the ashes from the rear to the front of the pit. Fig. 73 is a sectional view of the Playford stoker attached to a water tube boiler.

In class two may be included stokers having the grates inclined more or less. In some varieties the grates incline from front to rear, while in others they are made to incline from the side walls towards the center line of the furnace.

In the Vicars mechanical stoker the grate bars are somewhat of the shape of the ordinary grate and lie in two tiers in a horizontal position. The lower or back tier next the bridge wall is stationary, and is placed there for the purpose of catching what coal is carried

over the ends of the upper or moving grate bars. These have a slow reciprocating motion which gradually moves the hot coke back towards the bridge wall. The coal is fed from a hopper into two compartments, from



Fig. 72. Mansfield Chain Grate Stoker, Showing How it Can Be Withdrawn from under Boiler.

which it is pushed by reciprocating plungers onto a coking plate and from thence it passes to the grate bars. The motion of these bars has several intermediate variations, from a state of rest to a movement of

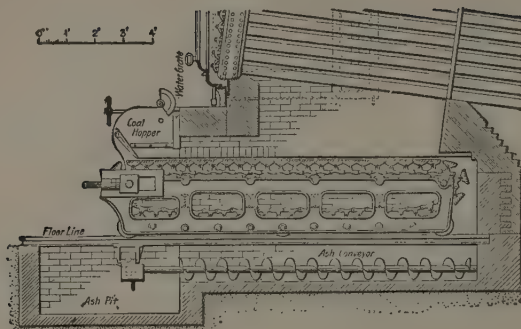


Fig. 73.

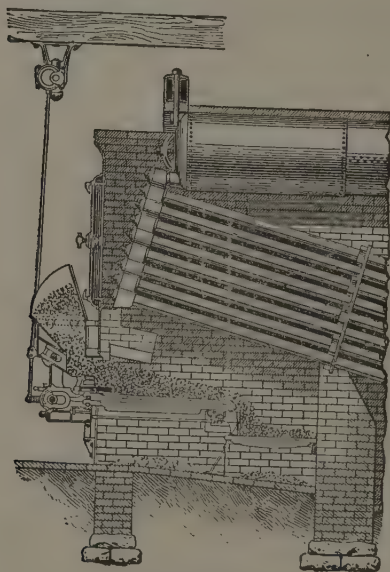


Fig. 74.

3½ in. They have a simultaneous movement forward by which the fuel is advanced, but on the return movement the bars act at separate intervals. In this manner the fuel remains undisturbed by the return motion of the grates. Fig. 74 illustrates this stoker.

In the Wilkinson stoker, Fig. 75, the grate bars are hollow and are set at an angle of 20°, the inclination being from front to back. Each bar is stepped along its fire surface and on the rise is perforated with a long, narrow slot. A steam pipe extends along the front of the furnace and from this pipe small branch pipes lead into the ends of the grate bars, which latter are in fact a series of hollow trunks with their front ends open. When in operation a steam blast is admitted to each of the several trunk grate bars through the small branch pipes, and this blast induces an air current of more or less pressure, which finds an outlet through the narrow slots in the stepped fire surface of the grates and directly into and through the burning mass of fuel. A slow reciprocating motion is imparted to the grates by means of cranks and links operated from an overhead shaft; see Fig. 76. These cranks are set alternately at 90° with each other, thus giving a forward movement to one-half of the grate bars while at the same time the other half is moving backward. In this manner the fuel is kept slowly moving down the inclined grates.

The Murphy Automatic Furnace, a sectional view of which is shown in Fig. 77, has the grates inclined inwards from the side walls, while a fire brick arch is sprung from side to side to cover the entire length of the grate. The coal is shoveled or fed by carrier into the coal magazines, one on each side, as shown in the

cut. If the furnace is placed directly under the boiler it necessitates putting these coal magazines within the



Fig. 75. Furnace View, Wilkinson Stoker.

side walls, but as the Murphy furnace is usually constructed at the present day as an outside furnace, the coal magazines are independent of the boiler walls.

The bottom of each magazine is used as a coking plate, against which the upper ends of the inclined grates rest. On the central part of this plate is an inverted

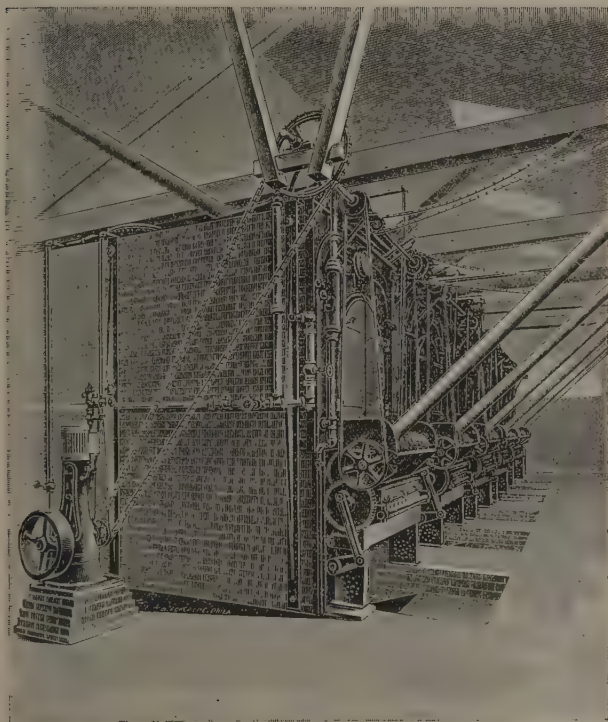


Fig. 76. Wilkinson Stoker.

open box. This is termed the "stoker box," and it is moved back and forth across the face of the coking plate by means of a shaft with pinions that mesh into racks

under each end of the box. By means of this motion of the stoker boxes the coal is pushed forward to the edge of the coking plate and from thence it slowly passes down over the inclined grates toward the center of the furnace. At this point the slowly rotating clinker breaker grinds the clinker and other refuse and deposits them in the ash pit.

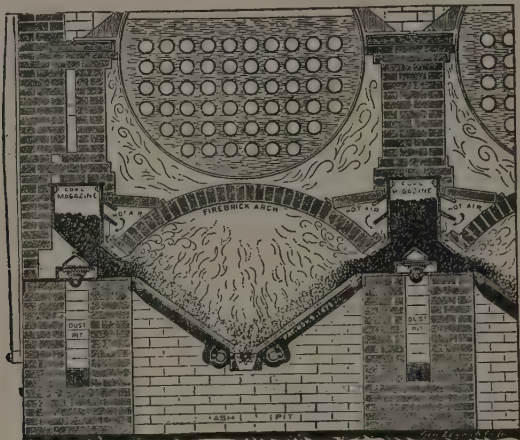


Fig. 77. The Murphy Automatic Furnace.

Above the coking plates are the "arch plates," upon which the bases of the fire brick arch rest. These plates are ribbed, the ribs being an inch apart, and, the arch resting upon these ribs, there is thus provided a series of air ducts by means of which the air, already heated by having been admitted in front and passed through the flues over the arch, is conducted into the furnace above the grates and comes directly in contact with the

gases rising from the coking fuel. Air is also admitted under the coking plate and, passing up through the grates, serves to keep them cool and also furnishes the needed supply to the burning coke as it slowly moves down toward the center.

The fuel is aided in its downward movement by the constant motion of the grates, one grate of each pair being moved up and down by a rocker at the lower end.

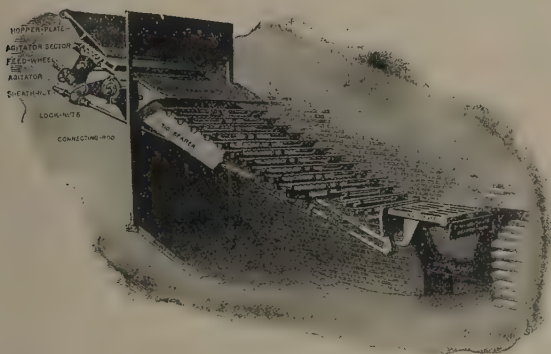


Fig. 78. Sectional Perspective of the Roney Mechanical Stoker.

Motion is imparted to the various moving parts of this furnace by means of a reciprocating bar extending across the outside of the entire front, and to which all the working parts are attached by links and levers. This bar is operated by a small engine at one side of the setting, the power required being about one horsepower per furnace.

The Roney stoker consists of a set of rocking stepped grate bars, inclined from the front towards the bridge wall. The angle of inclination is 37° . A dumping

grate operated by hand is at the bottom of the incline for the purpose of receiving and discharging the clinker and ash. This dumping grate is divided into sections for convenience in handling.

The coal is fed onto the inclined grates from a hopper in front. The grate bars rock through an arc of 30° , assuming alternately the stepped and the inclined position. Fig. 78 is a sectional perspective view of this stoker and illustrates the working parts.

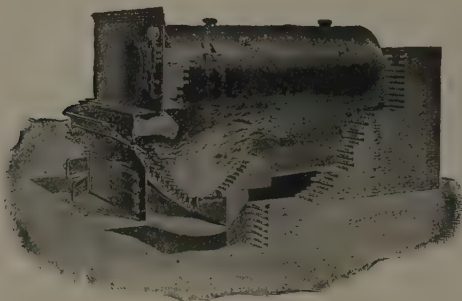


Fig. 79.

The grate bars receive their motion through the medium of a rocker bar and connecting rod. A shaft extending across the front of the stoker under the coal hopper carries an eccentric that gives motion to the connecting rod and also to the pusher in the coal hopper. This pusher, working back and forth, feeds the coal over the dead plates onto the grates, and its range of motion is regulated by a feed wheel from no stroke, to full stroke, according to the demand for coal. The motion of the grate bars may also be regulated by a sheath nut working on a long thread on the connecting

rod. Each grate bar consists of two parts, viz., a cast iron web fitted with trunnions on each end that rest in seats in the side bearer and a fuel plate having the under side ribbed to allow a free circulation of air.

The fuel plate is bolted to the web and carries the fuel. The grates lie in a horizontal position across the furnace in the form of steps, and ample provision is made for the admission of air through the slotted webs. A fire brick arch is also sprung across the furnace, covering the upper portion of the grate.

This arch, being heated to a high temperature, serves in a measure to partly coke the coal as it passes under it. Air is also admitted on top of the coal at the front. This air is heated by its passage through a perforated tile over the dead plate and adjoining the fire brick arch. Fig. 79 shows the location of the arch and tile.

In mechanical stokers of the under-feed type the air is supplied by forced draft.

The American stoker consists of a horizontal conveyor pipe into which the coal is fed from a hopper. The diameter of this pipe depends upon the quantity of coal to be burned, and varies from $4\frac{1}{2}$ in. for the smaller sizes up to 9 and 10 in. for the larger sized stokers. The length of the conveyor pipe for the standard 10-in. stoker is 72 in. Attached to the outer end of this conveyor pipe, and forming a part of it, is an iron box containing a reciprocating steam motor, which, through the medium of a rocker arm and pawl and ratchet wheel, drives a screw conveyor shaft that slowly revolves within the pipe, thus forcing the coal forward and up through another box or trough, which latter is wholly within the furnace. Extending around the top edges of this box, and on a level with the grate bars,

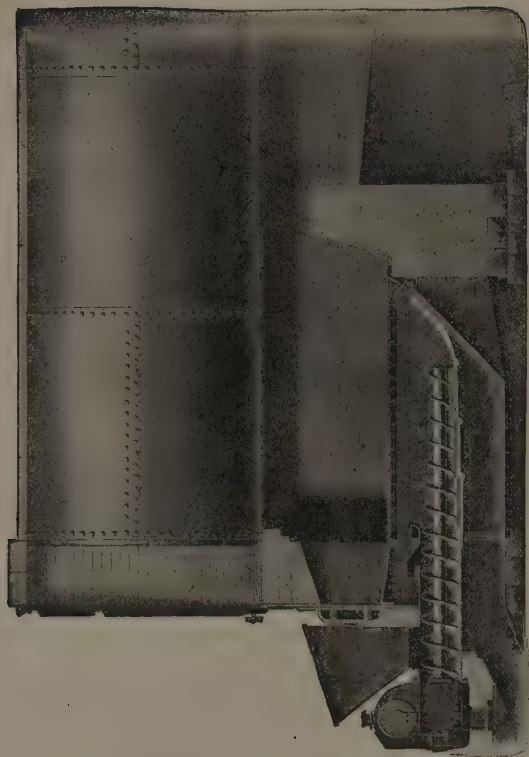


Fig. 80. Typical Setting of American Stoker under a Return Tubular Boiler.

there is a series of tuyeres through which the air is forced.

These tuyeres, being at a high temperature, serve to heat the air in its passage through them, thus greatly aiding combustion. Fig. 80 is a longitudinal sectional view of this stoker.

The speed of the screw conveyor is regulated by the hand throttle of the motor, according to the demand for coal. With the 9-in. standard stoker from 350 to 1,200

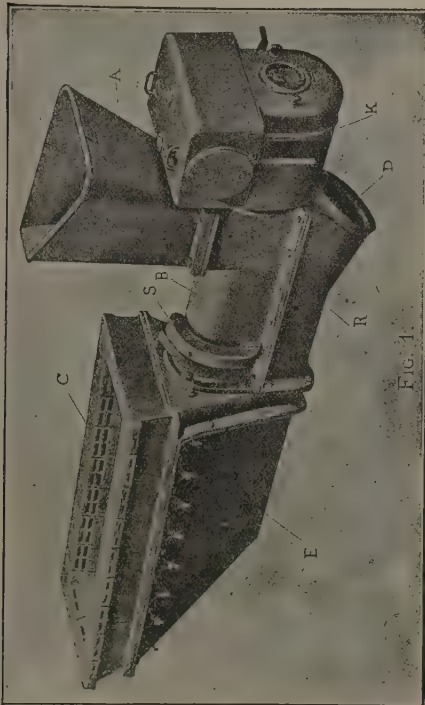


Fig. 81. Standard 8 and 9 inch American Stokers.

- | | |
|---|---|
| A. Coal hopper. | K. Automatic steam motor for driving conveyor. |
| B. Conveyor pipe. | R. Air pipe to wind-box. |
| C. Tuyeres for introducing air to the fuel. | S. Gas ducts for returning volatile products from entering coal to furnace. |
| D. Opening to wind-box for air connections. | |
| E. Wind-box for supplying air to tuyeres. | |

lbs. of coal per hour may be burned. Fig. 81 is a view of the American stoker before being placed in position in the furnace.

The air jets, passing out from the tuyeres in a horizontal direction, and from opposite sides, cut through the rounded bed of coal and the gases are thus ignited and consumed immediately after being distilled from the coal, while the pressure of the coal rising from underneath forces the already coked fuel over the edges of the trough or box onto the grates which occupy the space between the side walls and the coal trough. The air is first delivered from the fan into the air box that surrounds the coal trough on three sides and from thence it passes to the tuyeres. If this stoker is properly handled very good results may be obtained by its use, but, like all other devices for burning coal under boilers, it is bad policy to endeavor to force it beyond its capacity.

In the Jones under-feed stoker the coal is pushed forward and up into the furnace through a cast iron retort or trough. The impelling force is a steam ram connected to the outer end of the retort, and the speed of the ram is regulated automatically by the steam pressure, or by hand as desired. The coal is supplied to the ram through a cast iron hopper having a capacity of 125 to 140 lbs.

Force draft is also employed in this stoker, the air being conducted from the fan or blower through galvanized iron pipes into the closed ash pit, which really forms an air box, as the space on either side of the retort that is usually occupied by grate bars is in this case covered by solid cast iron dead plates, upon which the coked fuel lies until it is consumed. These plates, being hot, serve to heat the air coming in contact with them in its passage to the cast iron tuyeres through which it passes to the bed of burning fuel in the retort.

Air entirely surrounds the retort on the sides and back end, and is at a constant pressure in the ash pit, but can only pass into the furnace through the tuyeres, the jets of air cutting through the rounded heap of incandescent fuel from opposite sides, and in a direction inclined upwards.

Coal is supplied to the hopper either by hand, or by mechanical means where the plant is fitted with coal-handling machinery. The opening through which it passes from the hopper to a position in front of the ram is 8x10 in. in size. Each charge of the steam ram carries forward 15 to 20 lbs. of coal. Connected to the



Fig. 82.

ram, and moving in conjunction with it is a long rod extending through the retort near the bottom. Upon this rod are carried shoes that act as auxiliary plungers and facilitate the movement of the coal.

Fig. 82 is a sectional view of the Jones stoker, showing the machine full of coal, with the ram ready to make a charge. Fig. 83 shows the stoker complete before being placed in the furnace.

It is claimed by the builders of under-feed stokers, and the claim appears to have good foundation, that by pushing the green coal up so as to meet the upper crust

of glowing fuel the gases on being distilled immediately come in contact with and are consumed by the burning mass, and the formation of smoke is thus prevented. Both the Jones under-feed and the American stokers have proved to be very successful in the burning of the cheaper bituminous coals of the West. One feature tending to commend them is the fact that practically all of the coal is utilized, there being no waste caused by the slack coal or fine screenings dropped through the grate bars into the ash pit unconsumed.

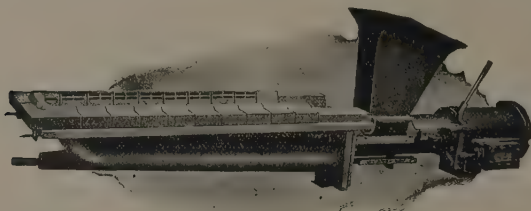
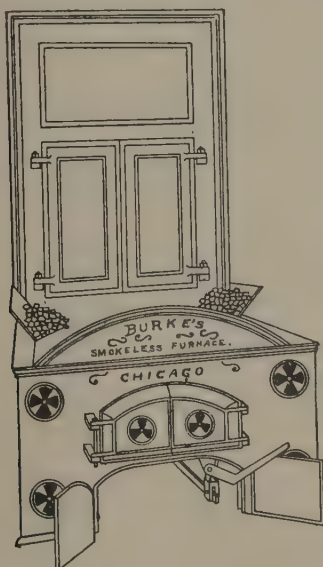


Fig. 83.

A good substitute for the mechanical stoker is an outside furnace, by which is meant a boiler installation having the furnace in front of instead of underneath the boiler. One of the principal hindrances to good combustion in the ordinary type of boiler furnace is the fact that the temperature of the boiler shell or water tubes with which the gaseous products of combustion come in contact can never be higher than the temperature of the water contained within the boiler. This temperature ranges from 297° for steam at 50 lbs. gauge pressure, up to 407° for 255 lbs. pressure, while the temperature of the furnace, according to Dr. Thurston and other high authorities, ranges from $2,010^{\circ}$ to $2,550^{\circ}$.

It is evident that perfect combustion does not take place until these high temperatures are reached. Each time the furnace is charged with fresh coal, especially if the boiler be hand-fired, a large volume of volatile gases is liberated, but not consumed. If these gases are

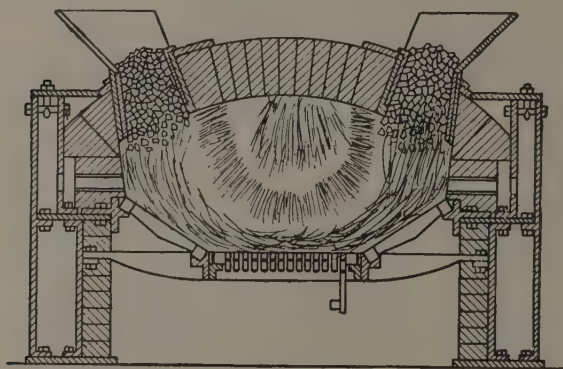


FRONT VIEW.

Fig. 84.

allowed to immediately come in contact with a comparatively cool surface, as for instance the heating surface of the boiler, the result is a cooling of the gases, incomplete combustion and the formation of smoke and soot. If, on the other hand, the furnace is so constructed that these gaseous products first impinge against hot

surfaces, such as fire brick arches or bafflers that have a temperature corresponding to that of the furnace, good combustion is assured. This condition is in a large degree attained by the use of outside furnaces that permit the construction of a fire brick arch to cover the entire grate surface.



CROSS SECTION.

Fig. 85.

The Burke furnace, patented by James V. Burke of Chicago, is a notable example of this type of furnace. It is applicable to any type of stationary boiler. Fig. 84 shows this furnace as applied to tubular boilers. It consists of a fire brick arch extending from 6 to 8 ft. outwards from the boiler front and of a width to correspond to the diameter of the boiler. The arch rests securely upon brick work inclosed in a well ventilated iron casing. There is practically no heat radiated from this furnace, all the heat generated by it passing to the boiler. The central portion of the grate bars consists

of shaking grates, while the side bars are stationary and inclined.

Fig. 85 is a sectional view and will serve to illustrate the construction of this furnace. The coal is fed through pockets on top on each side of the arch, the larger furnaces having two pockets on each side and the smaller sizes one. The doors in front are only opened for the purpose of cleaning fires or when first starting fires. The air is supplied by way of the ash pit, passing up through the grate bars. A portion of the air supply is also drawn through the ventilators and passes to the upper part of the furnace. The arch extends under the front end of the boiler 6 or 8 in., and there is a bridge wall about 4 ft. back from the front, against which the gases from the furnace impinge.

There are 42 sq. ft. of grate surface in the larger sizes, and 22 sq. ft. in the smaller size. Good combustion is attained in this furnace, owing to the fact that the gases as they are distilled from the coal come immediately in contact with the highly heated surface of the arch directly over the fire.

Stoker selection and stoker installation furnish problems requiring a high order of mechanical skill, knowledge of many details and freedom from bias and prejudice. Each installation must be studied individually and the size and type of stoker and subsidiary details of grate, draft, furnace, etc., determined only after an exhaustive consideration of all points. Thus and only thus may the highest economy be realized, for it is economy primarily that must be sought and secured in this day of keen competition, with a smokeless stack as a secondary consideration.

It is a fact, well recognized by all students of boiler

furnace combustion, that smoke suppression does not necessarily mean fuel economy, while it is universally recognized that when combustion is complete with fuel economy at its highest, there can be no smoke.

The very considerable initial cost of installing mechanical stokers may no longer be regarded as a species of speculation, but as a sound investment that returns a truly marvelous annual dividend. It is not overstating the facts as well recognized by progressive and studious engineers that there is no other single piece of power plant equipment that will pay a richer dividend on its cost than a properly constructed, properly selected, properly installed mechanical stoker.

It is totally impossible to duplicate the conditions of a stoker-fired furnace with one in which the coal is fed by hand through the firedoor.

CHIMNEYS.

Chimneys are required for two purposes—first, to carry off obnoxious gases: second, to produce a draft, and so facilitate combustion. The first requires size, the second height.

The weight of gas to be carried off by a chimney in a given time depends upon three things—size of chimney, velocity of flow, and density of gas. But as the density decreases directly as the absolute temperature, while the velocity increases, with a given height, nearly as the square root of the temperature, it follows that there is a temperature at which the weight of gas delivered is a maximum. This is about 550° above the surrounding air. Temperature, however, makes so little difference, that at 550° above, the quantity is *only four per cent* greater than at 300° . Therefore, height and area are the only elements necessary to consider in an ordinary chimney.

The intensity of draft is, however, independent of the size, and depends upon the difference in weight of the outside and inside columns of air, which varies nearly as the product of the height into the difference of temperature. This is usually stated in an equivalent column of water, and may vary from 0 to possibly 2 inches.

After a height has been reached to produce draft of sufficient intensity to burn fine, hard coal, provided the area of the chimney is large enough, there seems no good mechanical reason for adding further to the height, whatever the size of the chimney required. Where cost is no consideration, there is no objection to building as high

as one pleases; but for the purely utilitarian purposes of steam making equally good results might be attained with a shorter chimney at much less cost.

The intensity of draft required varies with the kind and condition of the fuel, and the thickness of the fires. Wood requires the least, and anthracite screenings the most. The strong draft required for burning the smaller sizes of anthracite coal necessitates a very tall chimney unless forced blast is used.

Generally a much less height than 100 feet cannot be recommended for a boiler, as the lower grades of fuel cannot be burned as they should be with a shorter chimney.

A round chimney is better than square, and a straight flue better than a tapering, though it may be either larger or smaller at the top without detriment.

The effective area of a chimney for a given power varies inversely as the square root of the height. The actual area, in practice, should be greater, because of retardation of velocity due to friction against the walls. On the basis that this is equal to a layer of air two inches thick over the whole interior surface, and that a commercial horse-power requires the consumption on an average of 5 pounds of coal per hour, we have the following formulas:

$$E = \frac{0.3 H}{\sqrt{h}} = A - 0.6 \sqrt{A} \dots\dots\dots 1$$

$$H = 3.33 E \sqrt{h} \dots\dots\dots 2$$

$$S = 12 \sqrt{E} + 4 \dots\dots\dots 3$$

$$D = 13.54 \sqrt{E} + 4 \dots\dots\dots 4$$

$$h = \left(\frac{0.3 H^2}{E} \right) \dots\dots\dots 5$$

in which H = horse-power; h = height of chimney in feet; E = effective area, and A = actual area in square feet; S = side of square chimney, and D = dia. of round chimney in inches. Table 12 is calculated by means of these formulas.

To find the draft of a given chimney in inches of water: Divide 7.6 by the absolute temperature of the external air ($\tau_a = t + 460$); divide 7.9 by the absolute temperature of the gases in the chimney ($\tau_c = t' + 460$); subtract the latter from the former, and multiply the remainder by the height of the chimney in feet. This rule, expressed in a formula, would be:

$$d = h \left(\frac{7.6}{\tau_a} - \frac{7.9}{\tau_c} \right).$$

To find the height of a chimney, to give a specific draft power, expressed in inches of water: Proceed as above, through the first two steps, then divide the given draft power by the remainder, the result is the height in feet. Or, by formula:

$$h = \frac{d}{\left(\frac{7.6}{\tau_a} - \frac{7.9}{\tau_c} \right)}$$

To find the maximum efficient draft for any given chimney, the heated column being 600° F., and the external air 62°: Multiply the height above grate in feet by .007, and the product is the draft power in inches of water.

The diagram, Fig. 86, shows the draft, in inches, of water for a chimney 100 feet high, under different temperatures, from 50° to 800° above external atmosphere, which is assumed at 60°. The vertical scale is full size, and each division is 1/20 of an inch. It also shows the

relative quantity, in pounds of air, which would be delivered, in the same time, by a chimney under the same differences of temperature. It will be seen that practi-

cally nothing can be gained by carrying the temperature of the chimney more than 350° above the external air at 60° .

To determine the quantity of air, in pounds, a given chimney will deliver per hour, multiply the distance in inches, at given temperature, on the diagram, Fig. 86, by 1,000 times the effective area in square feet, and by the square root of the height in feet. This gives a maximum. Friction in flues and furnace may reduce it greatly.

The external diameter of a brick chimney at the base should be one-tenth the height, unless it be supported by some other structure. The "batter" or taper of a chimney should be from $\frac{1}{16}$ to $\frac{1}{4}$ inch to the foot on each side.

Thickness of brick work: one brick (8 or 9 inches) for 25 ft. from the top, increasing $\frac{1}{2}$ brick (4 or $4\frac{1}{2}$ inches) for each 25 ft. from the top downwards.

If the inside diameter exceed 5 ft. the top length should be $1\frac{1}{2}$ bricks, and if under 3 ft. it may be $\frac{1}{2}$ brick for ten feet.

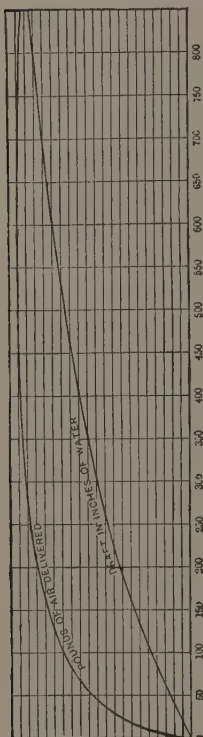


Fig. 86.

TABLE 11.

THEORETICAL DRAFT PRESSURE IN INCHES
OF WATER* IN A CHIMNEY 100 FEET HIGH.(For other heights the draft varies directly as the
height.)

Temp. in Chimney Fahr.	TEMP. OF EXTERNAL AIR. (BAROMETER 30 INCHES.)									
	0°	10°	20°	30°	40°	50°	60°	70°	80°	90° 100°
200°	.453	.419	.384	.353	.321	.292	.263	.234	.209	.182 .157
220	.488	.453	.419	.388	.356	.326	.298	.269	.244	.217 .192
240	.520	.488	.451	.421	.388	.359	.330	.301	.276	.250 .225
260	.555	.528	.484	.453	.420	.392	.363	.334	.309	.282 .257
280	.584	.549	.515	.482	.451	.422	.394	.365	.340	.313 .288
300	.611	.576	.541	.511	.478	.449	.420	.392	.367	.340 .315
320	.637	.603	.568	.538	.505	.476	.447	.419	.394	.367 .342
340	.662	.628	.598	.563	.530	.501	.472	.443	.419	.392 .367
360	.687	.653	.618	.588	.555	.526	.497	.468	.444	.417 .392
380	.710	.676	.641	.611	.578	.549	.520	.492	.467	.440 .415
400	.732	.697	.662	.632	.598	.570	.541	.513	.488	.461 .436
420	.753	.718	.684	.653	.620	.591	.563	.534	.509	.482 .457
440	.774	.739	.705	.674	.641	.612	.584	.555	.530	.503 .478
460	.793	.758	.724	.694	.660	.632	.603	.574	.549	.522 .497
480	.810	.776	.741	.710	.678	.649	.620	.591	.566	.540 .515
500	.829	.791	.760	.730	.697	.669	.639	.610	.586	.559 .534

*The available draft will be the tabular values less the amount consumed by friction in the stack. In stacks whose diameter is determined by the formulæ the net draft will be 80% of the tabular values. Hence to obtain from the table the height of stack necessary to produce a net draft of say 0.6 inches, the theoretical draft will be $0.6 \times 1.25 = 0.75$ inches, which can be got with a stack 100 ft. high with flue-gas temperature of 420° F., and air temperature of 0° F., or a stack 125 ft. high when the air temperature is 60° F.

TABLE 12.
SIZES OF CHIMNEYS WITH APPROPRIATE HORSE-POWER OF BOILERS.

Diameter in	HEIGHT OF CHIMNEYS AND COMMERCIAL HORSE-POWER.										Side of Square, Inches.	Effective Area, Square Feet.	Actual Area, Square Feet.
	50 ft.	60 ft.	70 ft.	80 ft.	90 ft.	100 ft.	110 ft.	125 ft.	150 ft.	175 ft.	200 ft.		
18	23	25	27	...	...	...	...	...	...	...	...	0.97	1.77
21	35	38	41	...	...	...	...	...	...	...	...	1.47	2.41
24	49	54	58	62	...	...	...	...	...	...	...	2.08	3.14
27	65	72	78	83	...	...	...	...	...	...	...	2.78	3.93
30	84	92	100	107	113	...	...	...	...	...	...	3.58	4.91
33	...	115	125	133	141	...	...	...	...	...	...	4.48	5.94
36	...	141	152	163	173	182	...	...	...	...	...	5.47	7.07
39	...	...	183	196	208	219	...	...	...	...	...	6.57	8.30
42	...	...	216	231	245	258	271	...	...	...	...	7.76	9.62
48	...	...	...	311	330	348	365	389	...	...	...	10.44	12.57
54	...	...	...	363	427	449	472	503	551	...	...	13.51	15.90
60	...	...	...	505	536	565	593	632	692	748	...	16.98	19.64
66	...	...	...	...	658	694	728	776	849	918	981	20.83	23.76
72	...	...	...	...	792	835	876	934	1023	1105	1181	25.08	28.27
78	...	...	...	...	...	995	1038	1107	1212	1310	1400	29.73	33.18
84	...	...	...	...	...	1163	1214	1294	1418	1531	1637	34.76	38.48
90	...	...	...	...	...	1344	1415	1496	1639	1770	1893	40.19	44.18
96	...	...	...	...	...	1537	1616	1720	1876	2027	2167	46.01	50.27
102	...	...	...	...	...	...	...	1946	2133	2303	2462	52.23	56.75
108	...	...	...	...	...	...	...	2192	2402	2594	2773	58.83	63.62
114	...	...	...	...	...	...	...	2459	2687	2903	3003	65.83	70.83
120	...	...	...	...	...	...	...	...	2990	3230	3452	73.22	78.54
126	...	...	...	...	...	...	...	...	3308	3573	3820	81.00	86.59
132	...	...	...	...	...	...	...	...	3642	3935	4205	89.19	95.03
138	...	...	...	...	...	...	...	...	3991	4311	4608	97.75	103.86
144	...	...	...	...	...	...	...	...	4357	4707	5031	106.72	113.10

IRON CHIMNEY STACKS.

In many places iron stacks are preferred to brick chimneys. Iron stacks require to be kept well painted to prevent rust, and generally, where not bolted down, they need to be braced by rods or wires to surrounding objects. With four such braces attached to an angle iron ring at $\frac{2}{3}$ the height of stack, and spreading laterally at least an equal distance, each brace should have an area in square inches equal to 1-1000 the exposed area of stack (diam. $\times$ height) in feet.

Stability, or power to withstand the overturning force of the highest winds, requires a proportionate relation between the weight, height, breadth of base, and exposed area of the chimney. This relation is expressed in the equation

$$C \frac{d h^2}{b} = W,$$

in which d = the average breadth of the shaft; h = its height; b = the breadth of base,—all in feet; W = weight of chimney in lbs., and C = a co-efficient of wind pressure per square foot of area. This varies with the cross-section of the chimney, and = 56 for a square, 35 for an octagon, and 28 for a round chimney. Thus a square chimney of average breadth of 8 feet, 10 feet wide at base and 100 feet high, would require to weigh $56 \times 8 \times 100 \times 10 = 448,000$ lbs. to withstand any gale likely to be experienced. Brickwork weighs from 100 to 130 lbs. per cubic foot, hence such a chimney must average 13 inches thick to be safe. A round stack could weigh half as much, or have less base.

AIR.

Pure Air is a mixture of oxygen and nitrogen in following proportions: by volume 20.91 parts oxygen to 79.09 parts nitrogen; by weight 23.15 parts oxygen to 76.85 parts nitrogen. Air in nature always contains other constituents such as dust, carbon dioxide, ammonia, ozone and water vapor.

Air being perfectly elastic, the density of the atmosphere decreases in geometrical ratio with the altitude. This fact has an important bearing on proportions of furnaces and stacks located in high altitudes, as will later appear. The atmospheric pressure for different altitudes is given in Table 17.

WEIGHT AND VOLUME OF AIR.

A cubic foot of air at 60° and under average atmospheric pressure, at sea level, weighs 536 grains, and 13.06 cubic feet weigh one pound. Air expands or contracts an equal amount with each degree of variation in temperature. Its weight and volume at any temperature under 30 inches of barometer may be found within less than one-half of one per cent by the following formula, in which W = weight in pounds of one cubic foot, V = volume in cubic feet, per pound, and τ = absolute temperature, or 460° added to that by the thermometer, = $t + 460$.

$$W = \frac{40}{\tau}$$

$$V = \frac{\pi}{40}$$

For any condition of pressure and temperature the following formulas are very nearly exact:

$$W = 2.71 \frac{p}{\tau}$$

$$V = \frac{\tau}{2.71p}$$

$$t = 2.71Vp - 460$$

in which p is pressure above absolute vacuum. The same formulæ answer for any other gas by changing the coefficient.

TABLE 13.

VOLUME AND WEIGHT OF AIR AT VARIOUS
TEMPERATURES, AND ATMOSPHERIC
PRESSURE.

Temperature in Degrees Fahr.	Volume of one Pound Cu. Ft.	Weight of one Cu. Ft. in Lbs.
50	12.840	.077884
55	12.964	.077133
60	13.090	.076400
65	13.216	.075667
70	13.342	.074950
75	13.467	.074260
80	13.593	.073565
85	13.718	.072894
90	13.845	.072230
95	13.970	.071580
100	14.096	.070942
110	14.346	.069698
120	14.598	.068500
130	14.849	.067342
140	15.100	.066221
150	15.352	.065140
160	15.603	.064088
170	15.854	.063072
180	16.106	.062090
190	16.357	.061134
200	16.606	.060210
210	16.860	.059313
212	16.910	.059135

TABLE 13—Continued.

Temperature in Degrees Fahr.	Volume of one Pound Cu. Ft.	Weight of one Cu. Ft. in Lbs.
220	17.111	.058442
230	17.362	.057596
240	17.612	.056774
250	17.865	.055975
260	18.116	.055200
270	18.367	.054444
280	18.621	.053710
290	18.870	.052994
300	19.121	.052297
320	19.624	.050959
340	20.126	.049686
360	20.630	.048476
380	21.131	.047323
400	21.634	.046223
425	22.262	.044920
450	22.890	.043686
475	23.518	.042520
500	24.146	.041414
525	24.775	.040364
550	25.403	.039365
575	26.031	.038415
600	26.659	.037510
650	27.913	.035822
700	29.172	.034280
750	30.428	.032865

COMBUSTION.

Combustion, as the term is used in steam engineering, is the rapid chemical combination of oxygen with carbon, hydrogen and sulphur, with the accompaniment of heat and light. The substance which combines with the oxygen is the *combustible*. The combustion is *perfect* when the combustible is oxidized to the highest possible degree; thus, conversion of carbon into carbon dioxide (CO_2) represents perfect combustion, while its conversion to monoxide (CO) is imperfect combustion, since the monoxide can be further burned and finally converted into CO_2 .

Kindling Point. As in many other chemical processes, a certain degree of heat is necessary to cause the union of the oxygen and combustible; the temperatures necessary to cause this union are the kindling temperatures, and are approximately as given in the following table by Stromeyer:

KINDLING TEMPERATURES.

Lignite Dust	300° F
Sulphur	470
Dried Peat	435
Anthracite Dust	570
Coal	600
Cokes	Red Heat
Anthracite	" " 750
Carbon Monoxide	" " 1211
Hydrogen	1030 or 1290

The Oxygen necessary for combustion is supplied from the air. Its density is 1.10521 (Air=1); its weight 0.088843 lbs. per cu. ft. at 32° F., and atmospheric pressure; its atomic weight is 16; a pound of air contains 0.2315 lbs. of oxygen, and one pound of oxygen is contained in 4.32 lbs. of air.

Carbon (C), the most abundant combustible, has atomic weight of 12, and reaches the boiler furnace as a constituent of oil, gas, coal, charcoal, wood, etc.

Hydrogen (H) occurs free in small quantity in some fuels, but is usually in combination with the carbon. Its atomic weight is 1; its density is 0.0692 (Air = 1); and its weight per cubic foot at 32° F. and atmospheric pressure is 0.00559 lbs. The heating value of one pound of pure carbon is rated at 14,500 heat units, while one pound of hydrogen gas contains 62,000 heat units.

COAL.

Analysis of coal shows that it contains moisture, fixed carbon, volatile matter, ash and sulphur in various proportions according to the quality of the coal. Table 14 deduced from a few of the many valuable tables of analysis of the coals of the United States will show the composition of the principal bituminous coals in use in this country for steam purposes. Two samples are selected from each of the great coal producing states, with the exception of Illinois, from which four were taken.

The process of combustion of fuel consists in the union of the carbon and hydrogen of the fuel with the oxygen of the air. Each atom of carbon combines with two atoms of oxygen, and the energetic vibration set up by

TABLE 14.

State	Kind of Coal	Moist- ure	Vola- tile Matter	Fixed Carbon	Ash	Sul- phur
Pennsylvania	Youghiogheny	1.03	36.49	59.05	2.61	0.81
"	Connellsville	1.26	30.10	59.61	8.23	0.78
West Virginia	Quinimont	0.76	18.65	79.26	1.11	0.23
"	Fire Creek	0.61	22.34	75.02	1.47	0.56
E. Kentucky	Peach Orchard	4.60	35.70	53.28	6.42	1.08
"	Pike County	1.80	26.80	67.60	3.80	0.97
Alabama	Cahaba	1.66	33.28	63.04	2.02	0.53
"	Pratt Co.'s	1.47	32.29	59.50	6.73	1.22
Ohio	Hocking Valley	6.59	35.77	49.64	8.00	1.59
"	Muskingum "	3.47	37.88	53.30	5.35	2.24
Indiana	Block	8.50	31.00	57.50	3.00	
"	"	2.50	44.75	51.25	1.50	
W. Kentucky	Nolin River	4.70	33.24	54.94	11.70	2.54
"	Ohio County	3.70	30.70	45.00	3.16	1.24
Illinois	Big Muddy	6.40	30.60	54.60	8.30	1.50
"	Wilmington	15.50	32.80	39.90	11.80	
"	" screenings	14.06	26.00	34.20	23.80	
"	Duquoin	8.90	23.50	60.60	7.00	

their combination is heat. Bituminous coal contains a large percentage of volatile matter which is released and flashes into flame when the coal is thrown into the furnace, and unless air is supplied in large amounts at this stage of the combustion there will be an excess of smoke and consequent loss of carbon. On the other hand there is a loss in admitting too much air because the surplus is heated to the temperature of the furnace without aiding the combustion and will carry off to the chimney just as many heat units as were required to raise it from the temperature at which it entered the furnace to that at which it enters the uptake. It will therefore be seen that a great advantage will be gained by first allowing the air that is needed above the fire to pass over or

through heated bridge walls or side walls. Some kinds of coal need more air for their combustion than do others, and good judgment and close observation are needed on the part of the fireman to properly regulate the supply.

Sulphur (S, atomic weight 32) is found in most coals and in some oils. It is usually present in a combined form, either as sulphide of iron, or sulphate of lime; in the latter form it has no heating value. Its presence in fuel is objectionable because the gases formed from its combustion attack the metal of the boiler and cause rapid corrosion, particularly in presence of moisture.

Nitrogen (N) is drawn into the furnace with the air. Its atomic weight is 14; its density is 0.9701 (Air = 1); its weight per cubic foot at 32° F. and atmospheric pressure is .07831 lbs.; each pound of air at atmospheric pressure contains 0.7685 lbs. of nitrogen, and one pound of nitrogen is contained in 1.301 lbs. of air.

Nitrogen performs no useful office in combustion, and passes through the furnace without change. It dilutes the air, absorbs heat and reduces the temperature of the products of combustion and is the chief source of heat loss in furnaces.

Combining Weights. When chemical elements unite to form a new compound they do so in definite proportions which are always the same, and the union produces heat, the quantity of which is also invariable. Thus, a pound of carbon, when carbon dioxide is formed, will always unite with $2\frac{2}{3}$ pounds of oxygen, and give off 14,600 B. T. U. As an intermediate step the carbon might unite with $1\frac{1}{3}$ times its weight of oxygen, and produce 4,450 B. T. U., but in its further conversion to CO_2 it would unite with an additional $1\frac{1}{3}$ times its weight of oxygen and evolve the other 10,150 B. T. U., since the

heat developed in any chemical combination depends upon the initial and final states, and not upon any intermediate change.

Calorific Value of Fuel. The amount of heat liberated per pound of fuel undergoing *perfect combustion* is called the calorific value of the fuel.

Some boilers will make steam more economically by partly closing the ash-pit doors, while others require the same doors to be kept wide open. The quantity of air required for the combustion of one pound of coal is, by volume, about 150 cu. ft.; by weight, about 12 lbs.

The temperature of the furnace is usually about 2500° , in some cases reaching as high as 3000° . The temperature of the escaping gases should not be much above nor below 400° F. for bituminous coal. The waste heat in the escaping gases can be utilized to great advantage by passing them through what are called economizers before they escape into the chimney. These economizers consist of coils or stacks of cast iron pipe placed within the flue or breeching leading from the boilers to the chimney and are enveloped in the hot gases, while the feed water is passed through the pipes on its way to the boilers, the result being that considerable heat is thus imparted to the feed water that would otherwise go to waste.

In order to attain the highest economy in the burning of coal in boiler furnaces two factors are indispensable, viz., a constant high furnace temperature, and quick combustion, and these factors can only be secured by supplying the fresh coal constantly just as fast as it is burned, and also by preventing as much as possible the admission of cold air to the furnace. This is why the automatic or mechanical stoker, if it be of the proper design, is more economical and causes less smoke than hand firing.

The fireman when he puts in a fire is prone to shovel in a good supply all at once, and this has the tendency to greatly reduce the temperature of the furnace, while at the same time it retards combustion. On the other hand the mechanical stoker supplies the coal continuously only as fast as it is required and no faster, and the furnace doors do not need to be opened at all, by which a large volume of cold air is prevented from entering the furnace and reducing the temperature. The author does not wish to be understood as recommending the adoption and use of mechanical stokers to replace hand firing, but he draws this contrast between the two methods of firing in order that it may be of some benefit to the thousands of honest toilers who earn a livelihood by shoveling coal into boiler furnaces.

The problem of the economical use of coal and the abatement of the smoke nuisance, especially in our large cities, has of late years become so serious that it is to the interest of every engineer, and especially every fireman, to use the utmost diligence, care and good judgment in the use of coal, and to emulate as much as possible the methods of the mechanical stoker.

Heat. All matter, whether solid, liquid or gaseous, consists of molecules, or atoms, which are in a state of continual vibration, and the result of this vibration is heat. The intensity of the heat evolved depends upon the degree of agitation to which the molecules are subject.

Heat Effects. When heat is added to or taken from a body, either the temperature of the body is altered, or its volume is varied, or its state is changed. Thus, if heat be added to water under atmospheric pressure, the temperature of the water increases until it reaches 212°F. If

more heat be added and the pressure remains unchanged. the temperature does not further increase, but the water evaporates into steam. Heat thus changes water from a liquid to a gaseous state. If heat be abstracted from water the temperature is reduced until it reaches 32° F., after which any diminution of heat does not further decrease the temperature, until the liquid is converted into a solid, or ice. The quantity of heat passing from one body to another can thus be estimated by the effects produced. Therefore heat is something that can be both transferred and measured.

The general effect of heat on a body is to increase its volume. If heat be abstracted from a body the contrary effect ensues, and the volume is diminished. Hence the general principle, to which, however, there are some exceptions, that heat expands and cold contracts. These effects, arising from a change of temperature, are produced in very different degrees according to the nature of the bodies. They are small in solids, greater in liquids, and greater still in gases.

It is well known that the work expended in friction apparently is lost as regards mechanical work; that heat is developed when friction occurs; that the greater the friction the greater is the amount of heat produced. Experiments have proved that the amount of heat generated by friction is exactly equivalent to the amount of work lost, whence it is shown that heat, like mechanical work is one of the forms of energy.

Thermometers. In consequence of the uniform expansion of mercury and its great sensitiveness to heat, it is the fluid most commonly used in the construction of thermometers. In all thermometers the freezing and the boiling point of water, under mean atmospheric pressure at

TABLE 15.

COMPARISON OF THERMOMETER SCALES.

	Fahrenheit	Centigrade	Reaumur
Absolute Zero.....	—460.66	—273.70	—218.96
	0	—17.77	—14.22
	10	—12.23	—9.77
	20	—6.67	—5.33
	30	—1.11	—0.88
Freezing Point.....	32	0.	0.
Maximum Density of } Water.....	39.1	3.94	3.15
	50	10.	8.
	75	23.89	19.11
	100	37.78	30.22
	200	93.34	74.66
Boiling Point.....	212	100.	80.
	250	121.11	96.88
	300	148.89	119.11
	350	176.67	141.33

$$F = \frac{9}{5}C + 32^\circ = \frac{9}{4}R + 32^\circ$$

$$C = \frac{5}{9}(F - 32^\circ) = \frac{5}{4}R.$$

$$R = \frac{4}{5}C = \frac{4}{9}(F - 32^\circ).$$

sea level, are assumed as two fixed points, but the division of the scale between these two points varies in different countries, hence there are in use three thermometers, known as the Fahrenheit, the Centigrade or Celsius, and the Réaumur. In the Fahrenheit the space between the two fixed points is divided into 180 parts; the boiling point is marked 212, and the freezing point is marked 32, and zero is a temperature which, at the time this thermometer was invented, was incorrectly imagined to be the lowest temperature attainable. In the Centigrade

and the Réaumur scales the distance between the two fixed points is divided into 100 and 80 parts, respectively. In each of these two scales the freezing point is marked 0 and the boiling point is marked 100 in the Centigrade and 80 in the Réaumur. Each of the 180, 100 or 80 divisions is termed a degree. Table 15 and the appended formulas are useful for converting one scale to another.

ABSOLUTE ZERO.

At 32° F. a perfect gas expands $\frac{1}{492.66}$ part of its volume, if its temperature is increased one degree and its pressure remains constant. This rate of expansion holds good at all temperatures above the freezing point, in the case of the gas, which would double its volume if under a constant pressure its temperature were raised to $32^{\circ} + 492.66 = 524.66^{\circ}$ F., while under a diminution of temperature it would shrink and finally disappear at a temperature of $492.66 - 32 = 460.66^{\circ}$ below zero F.

Therefore the temperature 460.66° , or for the sake of simplicity, 461° F. is taken as absolute zero.

Until as late as the beginning of the nineteenth century two rival theories in regard to the nature of heat had been advocated by scientists. The older of these theories was that heat was a material substance, a subtle, elastic fluid termed caloric, and that this fluid penetrated matter something like water penetrates a sponge. But this theory was shown to be false by the wonderful researches and experiments of Count Rumford at Munich, Bavaria, in 1798.

By means of the friction between two heavy metallic bodies placed in a wooden trough filled with water, one of the pieces of metal being rotated by machinery driven by horses, Count Rumford succeeded in raising the tem-

perature of the water in two and one-half hours from its original temperature of 60° to 212° F., the boiling point, thus demonstrating that heat is not a material substance, but that it is due to vibration or motion, an internal commotion among the molecules of matter. This theory, known as the Kinetic theory of heat, has since been generally accepted, although it was nearly fifty years after Rumford advocated it in a paper read before the Royal Society of Great Britain in 1798, before scientists generally became converted to this idea of the nature of heat, and the science of thermo-dynamics placed on a firm basis.

During the period from 1840 to 1849 Dr. Joule made a series of experiments which not only confirmed the truth of Count Rumford's theory that heat was not a material substance, but a form of energy which may be applied to or taken away from bodies, but Joule's experiments also established a method of estimating in mechanical units or foot pounds the amount of that energy. This latter was a most important discovery, because by means of it the exact relation between heat and work can be accurately measured.

The first law of thermo-dynamics is this: Heat and mechanical energy or work are mutually convertible. That is, a certain amount of work will produce a certain amount of heat, and the heat thus produced is capable of producing by its disappearance a fixed amount of mechanical energy if rightly applied. The mechanical energy in the form of heat which, through the medium of the steam engine, has revolutionized the world, was first stored up by the sun's heat millions of years ago in the coal which in turn, by combustion, is made to release it for purposes of mechanical work.

The general principles of Dr. Joule's device for measuring the amount of work in heat are illustrated in Fig. 87. It consisted of a small copper cylinder containing a known quantity of water at a known temperature. Inside the cylinder and extending through the top was a

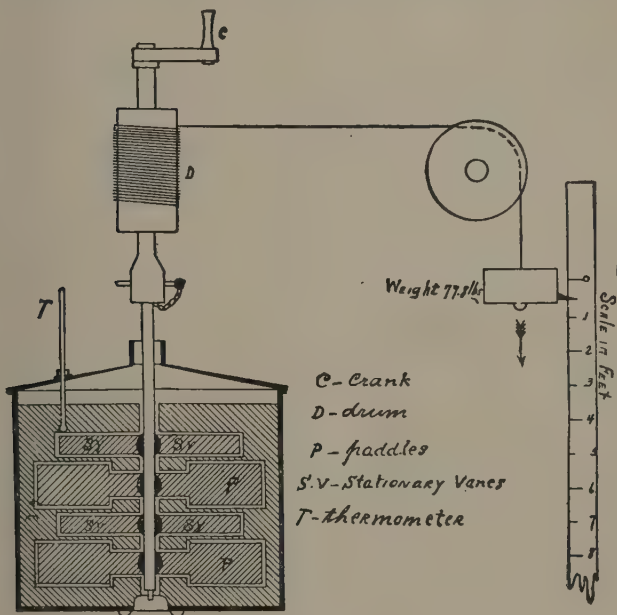


Fig. 87.

vertical shaft to which were fixed paddles for stirring the water. Stationary vanes were also placed inside the cylinder. Motion was imparted to the shaft through the medium of a cord or small rope coiled around a drum near the top of the shaft and running over a grooved

pulley or sheave. To the free end of the cord a known weight was attached. This weight was allowed to fall through a certain distance, and in falling it turned the shaft with its paddles, which in turn agitated the water, thus producing a certain amount of heat. To illustrate, suppose the weight to be 77.8 lbs., and that by means of the crank at the top end of the shaft it has been raised to the zero mark at the top of the scale. (See Fig. 87.) One pound of water at 39.1° F. is poured into the copper cylinder, which is then closed and the weight released. At the moment the weight passes the 10 ft. mark on the scale, the thermometer attached to the cylinder will indicate that the temperature of the water has been raised one degree. Then multiplying the number of pounds in the weight by the distance in feet through which it fell will give the number of foot pounds of work done. Thus, $77.8 \text{ lbs.} \times 10 \text{ ft.} = 778 \text{ foot pounds.}$

The heat unit or British thermal unit (B. T. U.) is the quantity of heat required to raise the temperature of one pound of water one degree, or from 39° to 40° F., and the amount of mechanical work required to produce a unit of heat is 778 foot pounds. Therefore the mechanical equivalent of heat is the energy required to raise 778 lbs. one foot high, or 77.8 lbs. 10 ft. high, or 1 lb. 778 feet high. Or again, suppose a one-pound weight falls through a space of 778 ft. or a weight of 778 lbs. falls one foot, enough mechanical energy would thus be developed to raise a pound of water one degree in temperature, provided all the energy so developed could be utilized in churning or stirring the water, as in Joule's machine. Hence the mechanical equivalent of heat is 778 foot pounds.

Specific Heat. The specific heat of any substance is

the ratio of the quantity of heat required to raise a given weight of that substance one degree in temperature, to the quantity of heat required to raise an equal weight of water one degree in temperature when the water is at its maximum density, 39.1° F. To illustrate, take the specific heat of lead, for instance, which is .031, while the specific heat of water is 1. That means that it would require 31 times as much heat to raise one pound of water one degree in temperature as it would to raise the temperature of a pound of lead one degree.

The following table gives the specific heat of different substances in which engineers are most generally interested:

Water at 39.1° F.....	1.000
Ice at 32° F.....	.504
Steam at 212° F.....	.480
Mercury	.033
Cast iron	.130
Wrought iron	.113
Soft steel	.116
Copper	.095
Lead	.031
Coal	.240
Air	.238
Hydrogen	3.404
Oxygen	.218
Nitrogen	.244

Sensible Heat and Latent Heat. The plainest and most simple definition of these two terms is that given by Sir Wm. Thomson. He says: "Heat given to a body and warming it is sensible heat. Heat given to a body and not warming it is latent heat." Sensible heat in a

substance is the heat that can be measured in degrees of a thermometer, while latent heat is the heat in any substance that is not shown by the thermometer.

To illustrate this more fully a brief reference to some experiments made by Professor Black in 1762 will no doubt make the matter plain. It will be remembered that at that early date comparatively little was known of the true nature of heat, hence Professor Black's investigations and discoveries along this line appear all the more wonderful. He procured equal weights of ice at 32° F. and water at the same temperature, that is, just at the freezing point, and placing them in separate glass vessels suspended the vessels in a room in which the uniform temperature was 47° F. He noticed that in one-half hour the water had increased 7° F. in temperature, but that twenty half hours elapsed before all of the ice was melted. Therefore he reasoned that twenty times more heat had entered the ice than had entered the water, because at the end of twenty half hours when the ice was all melted the water in both vessels was of the same temperature. The water having absorbed 7° of heat during the first half hour must have continued to absorb heat at the same rate during the whole of the twenty half hours, although the thermometer did not indicate it. From this he calculated that $7^{\circ} \times 20 = 140^{\circ}$ of heat had become latent or hidden in the water.

In another experiment Professor Black placed a lump of melting ice, which he estimated to be at a temperature of 33° F. on the surface, in a vessel containing the same weight of water at 176° F., and he observed that when the whole of the ice had been melted the temperature of the water was 33° F., thus proving that 143° of heat ($176^{\circ} - 33^{\circ}$) had been absorbed in melting the ice and

was at that moment latent in the water. By these two experiments Professor Black established the theory of the latent heat of water, and his estimate was very near the truth because the results obtained since that time by the greatest experimenters show that the latent heat of water is 142 heat units, or B. T. U.

Black's experiment for ascertaining the latent heat in steam at atmospheric pressure was made in the following simple manner: He placed a flat, open tin dish on a hot plate over a fire and into the dish he put a small quantity of water at 50° F. In four minutes the water began to boil, and in twenty minutes more it had all evaporated. In the first four minutes the temperature had increased $212^{\circ} - 50^{\circ} = 162^{\circ}$, and the temperature remained at 212° throughout the twenty minutes that it required to evaporate all the water, despite the fact that the water had been receiving heat during this period at the same rate as during the first four minutes. He therefore reasoned that in the twenty minutes the water had absorbed five times as much heat as it had in the four minutes, or $160^{\circ} \times 5 = 810^{\circ}$, without any sensible rise in temperature. Therefore the 810° became latent in the steam. Owing to the crude nature of the experiment Professor Black's estimate of the number of degrees of latent heat in steam was incorrect, as it has been proven by many famous experimenters since then that the latent heat of steam at atmospheric pressure is 965.7 B. T. U.

It will thus be perceived that what is meant by the term latent heat is that quantity of heat which becomes hidden or latent when the state of a body is changed from a solid to a liquid, as in the case of melting ice, or from a liquid to a gaseous state, as with water evaporated into steam. But the heat so disappearing has not been lost,

on the contrary it has, while becoming latent, been doing an immense amount of work, as can easily be ascertained by means of a few simple figures. It has been seen that a heat unit is the quantity of heat required to raise one pound of water one degree in temperature and also that the mechanical equivalent of heat, or, in other words, the mechanical energy stored in one heat unit is equal to 778 foot pounds of work.

A horse power equals 33,000 ft. lbs. of energy in one minute of time, and a heat unit $= 778 \div 33,000 = .0236$, or about 1/43 of a horse power. The work done by the heat which becomes latent in converting one pound of ice at 32° F. into water at the same temperature $= 142$ heat units $\times 778$ ft. lbs. $= 110,476$ ft. lbs., which divided by 33,000 equals 3.34 horse power. Again, by the evaporation of one pound of water from 32° F. into steam at atmospheric pressure, 965.7 units of heat become latent in the steam and the work done $= 965.7 \times 778 = 751,314$ ft. lbs. $= 22.7$ horse power. It will thus be seen what tremendous energy lies stored in one pound of coal, which contains from 12,000 to 14,500 heat units, provided all the heat could be utilized in an engine.

Total Heat of Evaporation. In order to raise the temperature of one pound of water from the freezing point, 32° F., to the boiling point, 212° F., there must be added to the temperature of the water $212^\circ - 32^\circ = 180^\circ$. This represents the sensible heat. Then to make the water boil at atmospheric pressure, or, in other words, to evaporate it, there must still be added 965.7 B. T. U., thus $180 + 965.7 = 1,145.7$, or in round numbers 1,146 heat units. This represents what is termed the total heat of evaporation at atmospheric pressure and is the sum of the sensible and latent heat in steam at that pressure.

But if a thermometer were held in steam evaporating into the open air, as, for instance, in front of the spout of a tea-kettle, it would indicate but 212° F.

When steam is generated at a higher pressure than 212° , the sensible heat increases and the latent heat decreases slowly, while at the same time the total heat of evaporation slowly increases as the pressure increases, but not in the same ratio. As, for instance, the total heat in steam at atmospheric pressure is 1,146 B. T. U., while the total heat in steam at 100 lbs. gauge pressure is 1,185 B. T. U., and the sensible temperature of steam at atmospheric pressure is 212° , while at 100 lbs. gauge pressure the temperature is 338 and the latent heat is 876 B. T. U.

Water. The elements that enter into the composition of pure water are the two gases, hydrogen and oxygen, in the following proportions:

By volume, hydrogen	2,	oxygen	1.
By weight, “	11.1,	“	88.9.

Perfectly pure water is not attainable, neither is it desirable nor necessary to the welfare of the human race, because the presence of certain proportions of air and ammonia add greatly to its value as an agent for manufacturing purposes and for generating steam. The nearest approach to pure water is rain water, but even this contains 2.5 volumes of air to each 100 volumes of water. Pure distilled water, such for instance as the return water from steam heating systems, is not desirable for use alone in a boiler, as it will cause corrosion and pitting of the sheets, but if it is mixed with other water before going into the boiler its use is highly beneficial, as it will prevent to a certain degree the formation of scale and

incrustation. Nearly all water used for the generation of steam in boilers contains more or less scale-forming matter, such as the carbonates of lime and magnesia, the sulphates of lime and magnesia, oxide of iron, silica and organic matter, which latter tends to cause foaming in boilers.

The carbonates of lime and magnesia are the chief causes of incrustation. The sulphate of lime forms a hard crystalline scale which is extremely difficult to remove when once formed on the sheets and tubes of boilers. Of late years the intelligent application of chemistry to the analyzing of feed waters has been of great benefit to engineers and steam users, in that it has enabled them to properly treat the water with solvents either before it is pumped into the boiler, or by the introduction into the boiler of certain scale preventing compounds made especially for treating the particular kind of water used. Where it is necessary to treat water in this manner great care and watchfulness should be exercised by the engineer in the selection and use of a boiler compound.

From ten to forty grains of mineral matter per gallon are held in solution by the waters of the different rivers, streams and lakes; well and mine water contains still more.

Water contracts and becomes denser in cooling until it reaches a temperature of 39.1° F., its point of greatest density. Below this temperature it expands and at 32° F. it becomes solid or freezes, and in the act of freezing it expands considerably, as every engineer who has had to deal with frozen water pipes can testify.

Water is 815 times heavier than atmospheric air. The weight of a cubic foot of water at 39.1° is approximately 62.5 lbs., although authorities differ on this matter, some

of them placing it at 62.379 lbs., and others at 62.425 lbs. per cubic foot. As its temperature increases its weight per cubic foot decreases until at 212° F. one cubic foot weighs 59.76 lbs.

The table which follows is compiled from various sources and gives the weight of a cubic foot of water at different temperatures.

TABLE 16.

Temper- ature	Weight per Cubic Foot	Temper- ature	Weight per Cubic Foot	Temper- ature	Weight per Cubic Foot
32° F.	62.42 lbs.	132° F.	61.52 lbs.	230° F.	59.37 lbs.
42°	62.42	142°	61.34	240°	59.10
52°	62.40	152°	61.14	250°	58.85
62°	62.36	162°	60.94	260°	58.52
72°	62.30	172°	60.73	270°	58.21
82°	62.21	182°	60.50	300°	57.26
92°	62.11	192°	60.27	330°	56.24
102°	62.00	202°	60.02	360°	55.16
112°	61.86	212°	59.76	390°	54.03
122°	61.70	220°	59.64	420°	52.86

The boiling point of water varies according to the pressure to which it is subject. In the open air at sea level the boiling point is 212° F. When confined in a boiler under steam pressure the boiling point of water depends upon the pressure and temperature of the steam, as, for instance, at 100 lbs. gauge pressure the temperature of the steam is 338° F., to which temperature the water must be raised before its molecules will separate and be converted into steam. In the absence of any pressure, as in a perfect vacuum, water boils at 32° F. temperature. In a vacuum of 28 in., corresponding to an absolute pressure of .943 lbs., water will boil at 100°, and in a vacuum of 26 in., at which the absolute

pressure is 2 lbs., the boiling point of water is 127° F. On the tops of high mountains in a rarefied atmosphere water will boil at a much lower temperature than at sea level, for instance at an altitude of 15,000 ft. above sea level water boils at 184° F.

Table 17 gives the boiling point of water at various altitudes above sea level, also the atmospheric pressure in pounds per square inch.

Steam. Having discussed to some extent the physical properties of water, it is now in order to devote some time to the study of the nature of steam, which is simply water in its gaseous form made so by the application of heat.

As has been stated in another portion of this book, matter consists of molecules or atoms inconceivably small in size, yet each having an individuality, and in the case of solids or liquids, each having a mutual cohesion or attraction for the other, and all being in a state of continual vibration more or less violent according to the temperature of the body.

The law of gravitation which holds the universe together, also exerts its wonderful influence on these atoms and causes them to hold together with more or less tenacity according to the nature of the substance. Thus it is much more difficult to chip off pieces of iron or granite than it is of wood. But in the case of water and other liquids the atoms, while they adhere to each other to a certain extent, still they are not so hard to separate, in fact, they are to some extent repulsive to each other, and unless confined within certain bounds the atoms will gradually scatter and spread out, and finally either be evaporated or sink out of sight in the earth's surface. Heat applied to any substance tends to accelerate the

TABLE 17.
BOILING POINT OF WATER AT VARIOUS
ALTITUDES.

Boiling Point in degrees Fahrenheit.	Altitude above Sea-Level. Feet.	Atmospheric Pressure. Pounds per square inch.	Barometer. Inches.
184	15,221	8.19	16.79
185	14,649	8.37	17.16
186	14,075	8.56	17.54
187	13,498	8.75	17.93
188	12,934	8.94	18.32
189	12,367	9.13	18.72
190	11,799	9.33	19.13
191	11,243	9.53	19.54
192	10,685	9.74	19.96
193	10,127	9.95	20.39
194	9,579	10.16	20.82
195	9,031	10.38	21.26
196	8,481	10.60	21.71
197	7,932	10.82	22.17
198	7,381	11.05	22.64
199	6,843	11.28	23.11
200	6,304	11.52	23.59
201	5,764	11.76	24.08
202	5,225	12.01	24.58
203	4,697	12.25	25.08
204	4,169	12.51	25.59
205	3,642	12.77	26.11
206	3,115	13.03	26.64
207	2,589	13.29	27.18
208	2,063	13.57	27.73
209	1,539	13.84	28.29
210	1,025	14.12	28.85
211	512	14.41	29.42
212	Sea-Level	14.70	30.00

vibrations of the molecules, and if enough heat is applied it will reduce the hardest substances to a liquid or gaseous state.

The process of the generation of steam from water is simply an increase of the natural vibrations of the molecules of the water, caused by the application of heat until they lose all attraction for each other and become instead entirely repulsive, and unless confined will fly off into space. But being confined they continually strike against the sides of the containing vessel, thus causing the pressure which steam or any other gas exerts when under confinement.

Of course steam, like other gases, when under pressure, is invisible, but the laws governing its action are well known. These laws, especially those relating to the expansion of steam, will be more fully discussed in the section on the Indicator. The temperature of steam in contact with the water from which it is generated, as for instance in the ordinary steam boiler, depends upon the pressure under which it is generated. Thus at atmospheric pressure its temperature is 212° F. If the vessel is closed and the pressure increased the temperature of the steam and also that of the water rises.

Saturated Steam. When steam is taken directly from the boiler to the engine without being superheated, it is termed saturated steam. This does not necessarily imply that it is wet and mixed with spray and moisture.

Superheated Steam. When steam is conducted into or through a vessel or coils of pipe separate from the boiler in which it was generated and is there heated to a higher temperature than that due to its pressure, it is said to be superheated.

Dry Steam. When steam contains no moisture it is

said to be dry. Dry steam may be either saturated or superheated.

Wet Steam. When steam contains mist or spray intermingled it is termed wet steam, although it may have the same temperature as dry saturated steam of the same pressure.

During the further consideration of steam in this book, saturated steam will be mainly under discussion for the reason that this is the normal condition of steam as used most generally in steam engines.

Total Heat of Steam. The total heat in steam includes the heat required to raise the temperature of the water from 32° F. to the temperature of the steam plus the heat required to evaporate the water at that temperature. This latter heat becomes latent in the steam, and is therefore called the latent heat of steam.

The work done by the heat acting within the mass of water and causing the molecules to rise to the surface is termed by scientists internal work, and the work done in compressing the steam already formed in the boiler or in pushing it against the superincumbent atmosphere, if the vessel be open, is termed external work. There are, therefore, in reality three elements to be taken into consideration in estimating the total heat of steam, but as the heat expended in doing external work is done within the mass itself it may, for practical purposes, be included in the general term latent heat of steam.

Density of Steam. The expression density of steam means the actual weight in pounds or fractions of a pound avoirdupois of a given volume of steam, as one cubic foot. This is a very important point for young engineers especially to remember, so as not to get the two

terms, pounds pressure and pounds weight, mixed, as some are prone to do.

Volume of Steam. By this term is meant the volume as expressed by the number of cubic feet in one pound weight of steam.

Relative Volume of Steam. This expression has reference to the number of volumes of steam produced from one volume of water. Thus the steam produced by the evaporation of one cubic foot of water from 39° F. into steam at atmospheric pressure will occupy a space of 1646 cu. ft., but, as the steam is compressed and the pressure allowed to rise, the relative volume of the steam becomes smaller, as for instance at 100 lbs. gauge pressure the steam produced from one cubic foot of water will occupy but 237.6 cu. ft., and if the same steam was compressed to 1,000 lbs. absolute or 985.3 lbs. gauge pressure it would then occupy only 30 cu. ft.

The condition of steam as regards its dryness may be approximately estimated by observing its appearance as it issues from a pet cock or other small opening into the atmosphere. Dry or nearly dry steam containing about 1 per cent of moisture will be transparent close to the orifice through which it issues, and even if it is of a grayish white color it may be estimated to contain not over 2 per cent of moisture.

Steam in its relation to the engine should be considered in the character of a vehicle for transferring the energy, created by the heat, from the boiler to the engine. For this reason all steam drums, headers and pipes should be thoroughly insulated in order to prevent, as much as possible, the loss of heat or energy by radiation.

There is a wide difference in the value of different substances for protection from radiation, their value

TABLE 18.

RELATIVE VALUE OF NON-CONDUCTING MATERIALS

SUBSTANCE.	VALUE.
*Loose Wool	3.35
*Loose Lampblack	1.12
*Geese Feathers	1.08
*Felt, Hair or Wool	1.
*Carded Cotton	1.
*Charcoal from Cork	.87
Mineral Wool	.68 to .83
Fossil Meal	.66 to .79
*Straw Rope, wound spirally	.77
*Rice Chaff, loose	.76
Carbonate Magnesia	.67 to .76
*Charcoal from Wood	.63 to .75
*Paper	.50 to .74
*Cork	.71
*Sawdust	.61 to .68
Paste of Fossil Meal and Hair	.63
Wood Ashes	.61
*Wood, across grain	.40 to .55
Loam, dry and open	.55
Chalk, ground, Spanish white	.51
Coal Ashes	.35 to .49
Gas-house Carbon	.47
Asbestos Paper	.47
Paste of Fossil Meal and Asbestos	.47
Asbestos, fibrous	.36
Plaster of Paris, dry	.34
Clay, with vegetable fiber	.34
Anthracite Coal, powdered	.29
Coke, in lumps	.27
Air Space, undivided	.14 to .22
Sand	.17
Baked Clay, Brick	.07
Glass	.05
Stone	.02

*Combustible, and sometimes dangerous.

varying nearly in the inverse ratio of their conducting power for heat, up to their ability to transmit as much heat as the surface of the pipe will radiate, after which they become detrimental, rather than useful, as covering. This point is reached nearly at baked clay or brick.

Table 18 shows the relative value of various non-conductors of heat, and table 19 gives the loss of heat from steam pipes protected and unprotected.

Where two values are given in table 18 for the same substance the lower one is for the denser condition.

A smooth or polished surface is of itself a good protection, polished tin or Russia iron having a ratio, for radiation, of 53 to 100 for cast iron. Mere color makes but little difference.

Hair or wool felt, and most of the better non-conductors, have the disadvantage of becoming soon charred from the heat of steam at high pressure, and sometimes of taking fire therefrom.

"Mineral wool," a fibrous material made from blast furnace slag, is the best non-combustible covering, but is quite brittle, and liable to fall to powder where much jarring exists.

Air space alone is one of the poorest of non-conductors, though the best owe their efficiency to the numerous minute air cells in their structure. This is best seen in the value of different forms of carbon, from cork charcoal to anthracite dust, the former being three times as valuable for this purpose, though in chemical constitution they are practically identical.

Any suitable substance used to prevent the escape of steam heat should not be less than one-inch thick.

The following table gives the loss of heat from steam pipes, naked and clothed with wool or hair felt, of dif-

TABLE 19.
LOSS OF HEAT FROM STEAM PIPES.

OUTSIDE DIAMETER OF PIPE, WITHOUT FELT.															
Thickness of Covering in inches.	2 inch diameter.			4 inch diameter.			6 inch diameter.			8 inch diameter.			12 inch diameter.		
	Loss in Units per foot run per hour.	Ratio of Loss.	Feet in Length per H. P. lost.	Loss in Units per foot run per hour.	Ratio of Loss.	Feet in Length per H. P. lost.	Loss in Units per foot run per hour.	Ratio of Loss.	Feet in Length per H. P. lost.	Loss in Units per foot run per hour.	Ratio of Loss.	Feet in Length per H. P. lost.	Loss in Units per foot run per hour.	Ratio of Loss.	Feet in Length per H. P. lost.
0	219.0	1.00	152	390.8	1.00	86	624.1	1.000	53	729.8	1.000	46	1077.4	1.000	31
$\frac{1}{4}$	100.7	.46	331	180.9	.46	182									
$\frac{1}{2}$	65.7	.30	507	117.2	.30	284	187.2	.300	177	219.6	.301	151	301.7	.280	114
1	43.8	.20	761	73.9	.18	451	111.0	.178	300	128.3	.176	259	185.3	.172	179
2	28.4	.13	1173	44.7	.11	745	66.2	.106	504	75.2	.103	443	98.0	.091	340
4	19.8	.09	1683	28.1	.07	1186	44.2	.066	808	46.0	.063	724	60.3	.056	553
6				23.4	.06	1424	33.7	.054	989	34.3	.047	972	45.2	.042	735

ferent thickness, the steam pressure being assumed at 75 lbs. and the external air at 60°.

FLOW OF STEAM THROUGH PIPES.

The approximate weight of any fluid which will flow in one minute through any given pipe with a given head or pressure may be found by the following formula:

$$W = 87 \sqrt{\frac{D(p_1 - p_2) d^5}{L \left(1 + \frac{3.6}{d}\right)}}$$

in which W = weight in pounds avoirdupois, d = diameter in inches, D = density or weight per cubic foot, p_1 = the initial pressure, p_2 = pressure at end of pipe, and L = the length in feet. Table 20 gives, approximately, the weight of steam per minute which will flow from various initial pressures, with one pound loss of pressure through straight smooth pipes, each having a length of 240 times its own diameter.

For sizes of pipe below 6-inch, the flow is calculated from the *actual* areas of "standard" pipe of such nominal diameters.

For horse-power, multiply the figures in the table by 2. For any other loss of pressure, multiply by the square root of the given loss. For any other length of pipe, *divide 240 by the given length expressed in diameters, and multiply the figures in the table by the square root of this quotient*, which will give the flow for 1 lb. loss of pressure. Conversely, dividing the given length by 240 will give the loss of pressure for the flow given in the table.

The loss of head due to getting up the velocity, to the friction of the steam entering the pipe, and passing el-

TABLE 20.
FLOW OF STEAM THROUGH PIPES.

DIAMETER OF PIPE, IN INCHES. LENGTH OF EACH=240 DIAMETERS.															
$\frac{3}{4}$		1	1 $\frac{1}{2}$	2	2 $\frac{1}{2}$	3	4	5	6	8	10	12	15	18	
Weight of Steam per minute, in pounds, with one pound loss of pressure.															
1	1.12	2.05	5.65	10.15	15.26	25.12	46.27	76.1	111.6	209.1	336.3	495.3	792	1160	
10	1.38	2.54	6.98	12.54	18.85	31.03	57.15	93.9	137.9	258.2	415.3	611.8	979	1433	
20	1.62	2.97	8.18	14.70	22.09	36.36	66.97	110.1	161.6	302.6	486.7	716.9	1147	1679	
30	1.82	3.34	9.21	16.54	24.86	40.92	75.37	123.9	181.8	340.6	547.8	806.9	1291	1889	
40	2.01	3.68	10.12	18.18	27.34	44.99	82.87	136.3	199.9	374.5	602.3	887.2	1419	2078	
50	2.17	3.98	10.95	19.67	29.57	48.67	89.64	147.4	206.3	404.9	651.5	959.7	1535	2248	
60	2.32	4.25	11.71	21.04	31.63	52.06	95.89	157.7	231.3	433.3	696.9	1026.5	1642	2404	
70	2.46	4.51	12.42	22.32	33.55	55.22	101.71	167.3	245.4	459.6	739.3	1088.9	1742	2550	
80	2.59	4.75	13.09	23.52	35.36	58.19	107.18	176.3	258.6	484.3	778.9	1147.4	1836	2687	
90	2.71	4.96	13.66	24.55	36.91	60.74	111.88	183.9	269.9	505.5	811.5	1197.8	1916	2805	
100	2.84	5.20	14.32	25.73	38.71	63.66	117.25	192.8	282.9	529.8	852.2	1255.3	2008	2940	
120	3.06	5.61	15.44	27.75	41.71	68.64	126.43	207.9	305.1	571.3	918.9	1353.6	2166	3170	
150	3.37	6.16	16.97	30.49	45.83	75.42	138.91	228.4	335.2	627.7	1009.6	1487.2	2379	3483	

bows and valves, will reduce the flow given in the tables. The resistance at the opening, and that at a globe valve, are each about the same as that for a length of pipe equal to 114 diameters divided by a number represented by $1 + (3.6 \div \text{diameter})$. For the sizes of pipes given in the table, these corresponding lengths are:

$\frac{3}{4}$	1	$1\frac{1}{2}$	2	$2\frac{1}{2}$	3	4	5	6	8	10	12	15	18
20	25	34	41	47	52	60	66	71	79	84	88	92	95

The resistance at an elbow is equal to $\frac{2}{3}$ that of a globe valve. These equivalents—for opening, for elbows, and for valves—must be added in each instance to the actual length of pipe. Thus a 4 in. pipe, 120 diameter (40 feet) long, with a globe valve and three elbows, would be equivalent to $120 + 60 + 60 + (3 \times 40) = 360$ diameters long; and $360 \div 240 = 1\frac{1}{2}$. It would therefore have $1\frac{1}{2}$ lbs. loss of pressure at the flow given in the table, or deliver $(1 \div \sqrt{1\frac{1}{2}} = .816)$ 81.6 per cent of the steam with the same (1 lb.) loss of pressure.

FLOW OF STEAM FROM A GIVEN ORIFICE.

Steam of any pressure flowing through an opening into any other pressure, less than three-fifths of the initial, has practically a constant velocity, 888 feet per second, or a little over ten miles per minute; hence the amount discharged in pounds is proportionate to the weight or density of the steam. To ascertain the pounds, avoirdupois, discharged per minute, *multiply the area of opening in inches, by 370 times the weight per cubic foot of the steam.* (See Table 10.)

Or the quantity discharged per minute may be approximately found by Rankine's formula:

$$W = 6 a p \div 7$$

in which W = weight in pounds, a = area in square inches, and p = absolute pressure. The theoretical flow requires to be multiplied by $k = 0.93$, for a short pipe, or 0.63 for a thin opening, as in a plate, or a safety valve.

Where the steam flows into a pressure more than $2/3$ the pressure in the boiler:

$$W = 1.9 a k \sqrt{(p - 8) 8}$$

in which 8 = difference in pressure between the two sides, in pounds per square inch, and a , p , and k as above.

To reduce to horse-power, multiply by 2.

Where a given horse-power is required to flow through a given opening, to determine the necessary difference in pressure:

$$8 = \frac{p}{2} - \sqrt{\frac{p^2}{4} - \frac{\text{H.P.}^2}{14a^2k}}$$

Evaporation Tests. The object of making evaporation tests of steam boilers is primarily to ascertain how many pounds of water the boilers are evaporating per pound of coal burned; but these tests can and should be made to determine several other important points with reference to the operation of the boilers, as for instance: 1. The efficiency of the boiler and furnace as an apparatus for the consumption of fuel and the evaporation of water; whether this apparatus is performing its guaranteed duty in this respect, and how it compares with a known standard. 2. To determine the relative economy of different varieties of coal, also to determine the relative value of fuels other than coal, such as oil, gas, etc

3. To ascertain whether or not the boilers as they are operated under ordinary every day conditions are being run as economically as they should be. 4. In case the boilers, owing to an increased demand for steam, fail to supply a sufficient quantity without forcing the fires, whether or not additional boilers are needed, or whether the trouble could be overcome by a change of conditions in operating them.

Tests for the last three purposes named can be made by the regular engineering force of the plant, but in case a controversy should arise between the maker of the boiler and the purchaser regarding the first mentioned point, viz., the guaranteed efficiency of the boiler or the furnace, the services of experts in boiler testing may be resorted to.

Preparing for a Test. All testing apparatus should be kept in such shape that it will not take three or four days to get it ready for making a test. On the contrary, it can be and should be always kept in condition ready for use, so that the preparations for making a test will occupy but a short time. A small platform scale sufficiently large for weighing a wheel-barrow load of coal should also be provided in addition to the apparatus heretofore described.

The capacity of each of the two tanks illustrated in Fig. 53, can be determined in two ways, either by measuring the cubical contents of each or by placing them one at a time on the scales, filling them with water to within a few inches of the top, and note the weight. Also make a permanent mark on the inside at the water level. The water should then be permitted to run out until within an inch or so of the outlet pipe near the bottom, where another plain mark should be made, after which the

empty tank should be again weighed, then by subtracting the last weight from the first the exact number of pounds of water that the tank will contain between the top and bottom marks can be determined and a note made of it.

It is much more convenient to have each tank contain the same quantity of water, although not absolutely necessary. The tanks should also be numbered 1 and 2 respectively in order to prevent confusion in keeping a record of the number of tanks full of water used during the test. Care should be exercised to have the water with which the tanks are filled while on the scale, at or near the same temperature as that at which it is to be fed into the boiler during the test. Otherwise there is liability of error owing to the variation in the weight of water at different temperatures. In order to guard against this, the capacity in cubic feet of each tank between the top and bottom marks should be ascertained by measuring the distance between the marks, also the diameter, or, if the tanks be square, the length of one side, after which the cubical contents can be easily figured and noted down. By knowing the capacity in cubic feet of each tank all possibility of error in the weight of feed water will be eliminated.

The scales for weighing the coal can be fitted with a temporary wooden platform large enough to accommodate a wheel-barrow, and after it has been balanced with the empty barrow on it, the record of weight of coal burned during the test can be easily kept.

The same barrow should be used throughout the test, and to save complications in estimating the weight, the same number of pounds of coal should be filled in the barrow each load. The coal passer will learn in a short time to fill the barrow to within a few pounds of the

same weight each load by counting the shovelful and the difference can easily be adjusted by having a small box of coal near the scale from which to take a few lumps to balance the load, or if there is too much coal in the barrow some of it can be thrown into the box.

At least two separate tally sheets should be provided, marked respectively coal and water, and the one for coal placed near the scale, and care should be taken that each load is tallied as soon as it is weighed. The tally sheet for water should be near the measuring tanks and as soon as a tank is emptied it should be tallied. The temperature of the feed water should be taken at least every thirty minutes, or oftener if possible, from a thermometer placed in the feed pipe near the check valve. The readings should be noted and, at the expiration of the test, the average taken.

If the object of the test is to ascertain the efficiency of the boiler and furnace it is absolutely necessary that the boiler and all its appurtenances be put in good condition, by cleaning the heating surface both inside the boiler and outside, scraping and blowing the soot out of the tubes if it be a return tubular boiler, and blowing the soot and ashes from between the tubes if it is a water tube boiler. All dust, soot and ashes should be removed from the outside of the shell and also from the combustion chamber and smoke connections. The grate bars and sides of the furnace should be cleared of all clinker, and all air leaks made as close as possible. The boiler and all its water connections should be free from leaks, especially the blow-off valve or cock. If any doubt exists as to the latter it should be plugged or a blind flange put on it. It is very essential that there should be no way for the water to leak out of the boiler, neither should

any water be allowed to get into the boiler during the test except that which is measured by passing through the tanks.

In making an efficiency test it is essential that the boiler should be run at its fullest capacity from the beginning to the end of the test. Therefore arrangements should be made to dispose of the steam as fast as it is generated. If the boiler is in a battery connected with a common header, its mates can be fired lighter during the test, but if there is but the one boiler in use a waste steam pipe should be temporarily connected through which the surplus steam, if there is any, can be discharged into the open air through a valve regulated as required. Before starting the test the boiler should be thoroughly heated by having been run several hours at the ordinary rate. The fire should then be cleaned and put in good condition to receive fresh coal.

At the time of beginning the test the water level should be at or near the height ordinarily carried and its position marked by tying a cord around one of the guard rods of the gauge glass, and, to prevent all possibility of error, the height of the water in the glass should be measured and a note made of it. Note also the time that the first lot of weighed coal is fed to the furnace and record it as the starting time. The steam pressure should be noted at the beginning of the test and at regular intervals during the progress of the test in order that the average pressure may be obtained.

At the close of the test all of the above conditions should be as nearly as possible the same as at the beginning; the quantity and condition of the fire should be the same, also the steam pressure and water level. This can

be accomplished only by careful work towards the close of the test.

During the progress of the test care should be exercised to prevent any waste of coal, especially in cleaning the fire. The ash made during the test must not be wet down until after it is weighed, as in all calculations for combustible and non-combustible matter in the coal the ash should be dry.

The duration of the test should be at least ten hours if it is possible to continue it for that length of time. The feed pump should be kept running at such speed as will supply the water to the boiler as fast as it is evaporated, and no faster. If at the close of the test a portion of water is left in the last tank tallied it can be measured and deducted from the total. And if any weighed coal is left on the floor it should be weighed back and deducted from the total weight. If the boiler is fed by an injector instead of a pump during the test, the injector should receive steam directly from the boiler under test through a well protected pipe. Also, the temperature of the feed water should be taken from the measuring tanks, or at least from the suction side of the injector, for the reason that the water in passing through the injector receives a large quantity of heat imparted to it by live steam directly from the boiler. Therefore the temperature of the water after it leaves the injector would not be a true factor for figuring the evaporation.

Determination of the Percentage of Moisture in the Steam. This is an important point in estimating the results of an evaporation test for the reason that each pound weight of moisture in the steam as it leaves the boiler represents a pound of water that has not been evaporated into steam, and should therefore be deducted

from the total weight of water fed into the boiler during the test.

The steam should be tested for moisture by taking samples of it from the steam pipe or header as near the boiler as possible in order to guard against additional moisture caused by condensation.

Practically all saturated steam contains water, varying in amount from a fraction of one per cent when the steam is generated in a properly designed boiler fed with good water, to five per cent or even more when the feed water is bad, or the boilers are of defective design. Not only is the heat absorbed by raising this water from the boiler feed temperature to the steam temperature practically wasted, but the water causes further loss by increasing the initial condensation in the engine cylinder; it also interferes with proper cylinder lubrication, causes knocking in the engine, and water hammer in the steam pipe.

Quality of Steam. The percentage weight of steam, in a mixture of steam and water, is called the quality of the steam. Thus steam of quality 99.5 contains one-half of one per cent by weight of moisture.

Calorimeters. The apparatus used to determine the moisture in steam is called a calorimeter, though the name is inapt, since the instrument is in no sense a measurer of heat. The first form used was the "barrel calorimeter," in this apparatus liability of error is so great that its use is practically abandoned. Modern calorimeters are usually of either the throttling or separator type.

Throttling Calorimeter. Fig. 88 shows a section through a typical form of the instrument. Steam is

drawn from the vertical pipe by a nipple arranged as later described, passes around the first thermometer cup as shown, then through a hole about $\frac{1}{8}$ -inch diameter in the disk as shown. It next passes around the lower thermometer cup, after which it is permitted to escape. Thermometers are inserted into the cups, which are then filled with cylinder oil, and when the whole apparatus is heated the temperature of the steam before, and after passing through the hole in the disk is noted.

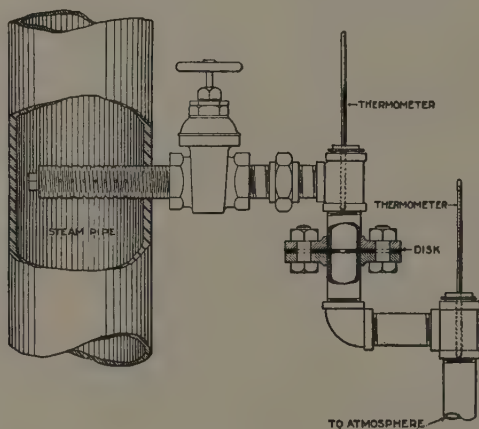


Fig. 88. Throttling Calorimeter and Sampling Pipe.

The instrument and pipes leading to it should be thoroughly covered to diminish the radiation loss.

When steam passes from a higher to a lower pressure, as in this case, no work has to be done in overcoming a resistance; hence assuming there is no loss from radiation, the quantity of heat is exactly the same after passing the disk as it was ahead of it. Suppose that the

higher steam pressure is 150 lbs. by gauge, and the lower pressure that of the atmosphere. The total heat in a pound of dry steam at the former pressure is 1193.5 B. T. U. and at the latter pressure is 1146.6 B. T. U., difference, 46.9 B. T. U. As this heat still exists in the steam of lower pressure, its effect is to *superheat* that steam. Assuming the specific heat of steam to be 0.48, the steam will then be superheated $\frac{46.9}{0.48} = 97.7$ degrees.

Suppose, however, the steam had contained one per cent. of moisture. Before any superheating could occur, this moisture would have to be evaporated into steam of atmospheric pressure. Since the latent heat of steam at atmospheric pressure is 965.8 B. T. U. it follows that the one per cent. of moisture would require 9.658 B. T. U. to evaporate it, leaving only $46.9 - 9.658 = 37.242$ B. T. U. available for superheating, hence the superheat would be $\frac{37.242}{0.48} = 77.6^\circ$ as against 97.7 degrees in the preceding case. In a similar manner the degree of superheat for other amounts of moisture can be determined, and the action of the throttling calorimeter is based on this fact as will now be shown.

Let H = total heat of steam at boiler pressure.

L = latent heat of steam at boiler pressure.

h = total heat of steam at reduced pressure after passing the disk.

t_1 = temperature of *saturated* steam at the reduced pressure.

t_2 = temperature of steam after expanding through opening in the disk.

0.48 = specific heat of saturated steam.

x = proportion of moisture in the steam.

The difference between the B. T. U.'s in a pound of steam at boiler pressure and after passing the disk is the heat which must evaporate the moisture in the steam, and then do the superheating, hence

$$H - h = xL - 0.48 (t_2 - t_1), \text{ therefore}$$

$$x = \frac{H - h - 0.48 (t_2 - t_1)}{L} \quad [6]$$

Almost invariably the lower pressure is taken as that of the atmosphere where $h = 1146.6$ and $t_1 = 212$, hence the formula becomes

$$x = \frac{H - 1146.6 - 0.48 (t_2 - 212)}{L} \quad [7]$$

For practical work it is more convenient to dispense with the upper thermometer in the calorimeter, and substitute an accurate steam gauge whose readings are more easily noted.

Sources of Error. There are two. The first is that the specific heat of superheated steam, while given as 0.48 is far from being certain, and only future investigation can determine the true value. The second source of error is loss of heat by radiation. Evidently from the moment the steam enters the sampling nipple it is losing heat, hence when it passes through the small opening and into the lower pressure the heat available for evaporating moisture, and superheating will be diminished by just the amount lost by radiation, hence the value of t_2 will be lower than it should be. This is sometimes corrected for as follows: A valve in the steam pipe beyond the calorimeter nipple is closed, and the steam left in a quiescent state for about ten minutes, and it is *assumed* that by

doing this all the moisture in the steam will settle out, and that a sample of steam drawn from the pipe will be dry. Steam is then allowed to flow through the calorimeter and the temperature of the lower thermometer is noted. Let T denote this temperature. Since the sample of steam was assumed to be dry it follows that if there were no loss from radiation the value of T would be that due to all of the liberated heat being absorbed in superheating the steam of lower temperature. There is, however, a loss by radiation, and the effect of this is to condense some of the steam of lower pressure, and the water thus formed must be evaporated before any superheating can be done. Let x^1 represent the proportion of water thus formed, then evidently

$$x^1 = \frac{H - h - 0.48 (T - t_1)}{L}$$

Now this amount of water was not in the steam originally, but was caused by condensation in the instrument, hence the *true* amount of moisture in the steam, which may be denoted by X , will be

$$\begin{aligned} X = x - x^1 &= \frac{H - h - 0.48 (t_2 - t_1)}{L} \\ &\quad - \frac{H - h - 0.48 (T - t_1)}{L} \\ &= \frac{0.48 (T - t_2)}{L} \end{aligned} \quad [8]$$

The disadvantages of this method are: (1) It assumes that during the test the boiler pressure will remain the same as it was when T was determined, which is seldom

practicable; (2) It assumes that the sample of steam drawn into the instrument when determining T was absolutely dry, although experiment has shown that this assumption is not necessarily true. Notwithstanding these facts, formula [8] is much used by engineers because of its simplicity and convenience, and any error due to its use is of no practical significance.

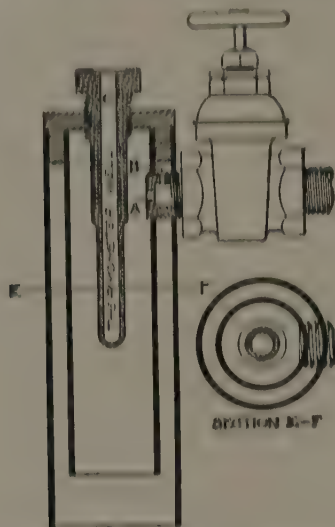


Fig. 89. Compact Throttling Calorimeter.

There are many forms of throttling calorimeter, all of which operate on precisely the same principle as the simple design shown in Fig. 88. An extremely convenient and compact design is shown in Fig. 89. It consists of two concentric cylinders screwed to a cap containing

a thermometer cup. The steam pressure is measured by a gauge placed in the supply pipe, or any other convenient place. Steam passes through the opening *A*, expands to atmospheric pressure, and its temperature at this pressure is measured by a thermometer placed in the cup *C*. To prevent radiation losses the annular space between the two cylinders is used as a jacket, and is supplied with steam through the hole *B*.

The limits of the throttling calorimeter at sea level are from about four per cent of moisture at eighty pounds pressure to six per cent at 200 pounds pressure. If there is a greater content of moisture the liberated heat is insufficient to evaporate it, and superheat the steam thus generated.

Separating Calorimeter. The separating calorimeter (Fig. 90) mechanically separates the entrained water from the steam and collects it in a reservoir, where its amount is either indicated by a gauge glass or determined by draining it off and weighing it. The steam passes out of the calorimeter through an orifice of known size, so that either its total amount can be calculated or it can be weighed as later described. To avoid radiation errors, the calorimeter should be well covered with non-conducting material. This instrument is not limited in capacity theoretically, but if the amount of moisture is very large, the readings should be checked by passing the discharged steam through a throttling calorimeter; that is, a small separator should be used between the steam pipe and a throttling calorimeter, and the sum of the percentages obtained from the two instruments be taken as the moisture in the steam.

In the separating calorimeter, the amount of steam

passing through the orifice can be determined by Napier's empirical formula,

$$\text{Pounds of steam per second} = \frac{pa}{70}$$

In which p = absolute pressure in pounds per square inch, and a = area of orifice in square inches.

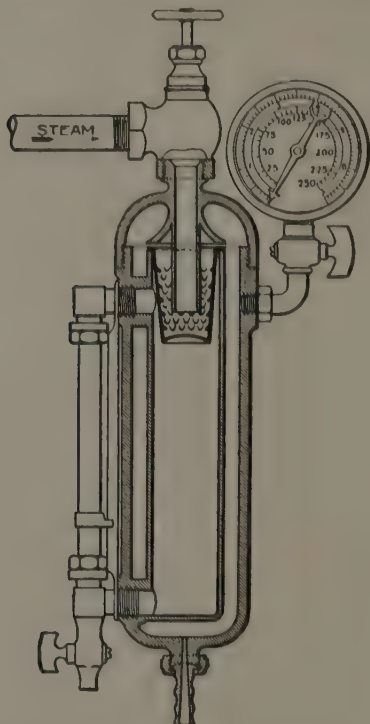


Fig. 90. Separating Calorimeter.

There is liability of considerable error in determining the area of such small orifices, and further, the flow of steam soon wears the orifice larger. A more accurate method of determining the weight of steam passing through is to convey it through a hose into a barrel of water resting on a platform scale. The weight of the barrel and contained water having been noted before and after the steam is run in, the difference is the weight of steam condensed. The moisture caught in the separating calorimeter can be weighed in the same way. If W is the weight of steam condensed, w the weight of moisture from the separating calorimeter, and x the per cent. of moisture in the steam, then

$$x = \frac{100w}{W + w} \quad [9]$$

Location of Sampling Nipple. The principal source of inaccuracy in calorimeter determinations is failure to secure an *average* sample of steam. It is extremely doubtful whether such a sample is ever secured. To diminish the liability of error the instrument should be located as near as possible to the point where the sample is drawn off.

Taking an Observation. Locate the sampling nipple as above directed, attach the instrument as close to it as possible, and cover all exposed parts to prevent radiation. If the throttling calorimeter be used, locate the steam gauge on the pressure side, and the thermometer on the expansion side. To take an observation, note simultaneously the gauge reading and the thermometer reading, and from these the content of moisture may be determined by use of formula [7]. If the separating calorimeter be used, attach to the separator

outlet a piece of hose which terminates in a vessel of water on a platform scale graduated to read to $1/100$ of a pound. Similarly connect the steam outlet to another vessel of water resting on an equally sensitive scale. Note in each case the weight of each vessel including the water it contains. When ready to take an observation, blow out the instrument thoroughly, so there will be no water in the separator. Then simultaneously close the separator drip and insert the steam hose into its vessel of water. When the separator has accumulated a sufficient quantity of water, close the valve at the main steam pipe, thus cutting off the supply of steam to the instrument, remove the steam hose from the vessel of water into which it was inserted, and empty the separator water into its vessel on the scale. Note the final weight of each vessel and contents, then the differences between final and original weights will be respectively, the weight of moisture collected by the separator, and the weight of steam from which the moisture was taken, hence the proportion of moisture can be computed from formula [9].

Before taking any calorimeter observations, steam should be allowed to flow through freely until the instrument is thoroughly heated up.

Moisture in the Coal. This can generally be obtained from the reports of the geologist of the state in which the coal is mined, or from the dealer, although the former is the most reliable. The percentage of moisture must be deducted from the total weight of coal in figuring the weight of combustible.

Measuring the Chimney Draft. A good draft is indispensable for obtaining economical results in an evaporation test. The draft can be easily regulated by a damper

to suit the conditions. Chimney draft is ordinarily measured by a draft gauge connected with the smoke flue near the chimney. The usual form of draft gauge is a glass tube bent in the shape of the letter U. (See Fig. 91.) One leg is connected to the flue by a small rubber

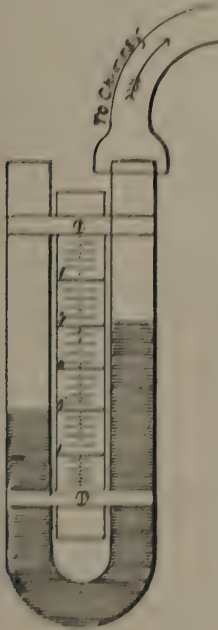


Fig. 91.

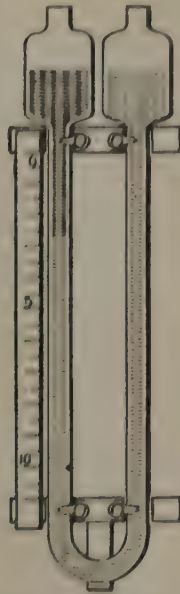


Fig. 92. Harriss' Draft Gauge.

hose, while the other is open to the atmosphere. The tube is partly filled with water, which will, when there is no draft, stand at the same height in both legs. When connected to the chimney or flue the suction will cause the

water in the leg to which the hose is attached to rise, while the level of the water in the other leg will be equally depressed, and the extent of the variation in fractions of an inch is the measure of the draft. Thus the draft is referred to as being .5, .7 or .75 inches. The draft should not be less than .5 inches in any case to insure good results.

The Barrus draft gauge is illustrated in Fig. 92. It consists of a U-tube made of $\frac{1}{2}$ -inch glass, surmounted by two larger tubes, or chambers, each having a diameter of $2\frac{1}{2}$ -inch. Two different liquids which will not mix,

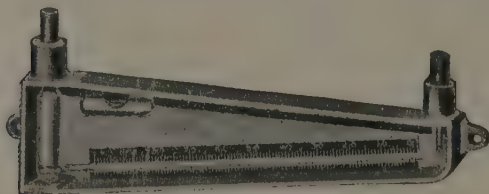


Fig. 93. Ellison's Draft Gauge Outline.

and which are of different color, are used. The movement of the line of demarcation is proportional to the difference in the areas of the chambers and of the U-tube connecting them below. The liquids generally employed are alcohol colored red and a certain grade of lubricating oil. A multiplication varying from eight to ten times is obtained under these circumstances; in other words, with $\frac{1}{4}$ -inch draft the movement of the line of demarcation is some 2 inches. The instrument is calibrated by referring it to the ordinary U-tube gauge.

Ellison's Gauge. In this form of gauge the lower portion of the ordinary U-tube has been replaced by a tube slightly inclined to the horizontal, as shown in Fig. 93.

By this arrangement any vertical motion in the left hand upright tube causes a very much greater travel of the liquid in the inclined tube, thus permitting extremely small variation in the draft pressure to be read with facility.

The gauge is first leveled by means of the small level attached to it, both legs being open to the atmosphere. The liquid is then adjusted (by adding to or taking from it) until its meniscus rests at the zero point on the right. The left hand leg is then connected to the source of draft by means of a piece of rubber tubing. Under these circumstances, a rise of level of one inch in the left hand vertical tube causes the meniscus in the inclined tube to pass from the point 0 to 1.0. The scale is divided into tenths of an inch, and the subdivisions are hundredths of an inch.

The right hand leg of the instrument bears two marks. By filling the tube to the lower of these the range of the instrument is increased one-half inch, i. e., it will record draft pressures from 0 to $1\frac{1}{2}$ inches. Similarly, by filling to the upper mark, the range is increased to 2 inches. When so used the observed readings in the scale are to be increased by one-half or one-inch, as the case may be.

The makers recommend the use of a non-drying oil for the liquid, usually a 300° test refined petroleum, but water suffices for all practical purposes.

Flue Gas Analysis. The object of the flue-gas analysis is to determine from a sample of the gas the amount of excess air admitted, the degree of completeness of the combustion of the carbon, and the amount and distribution of the heat losses due to the excess air and incomplete combustion. The quantities actually determined by the analysis are the relative proportions of carbon

dioxide (CO_2), carbon monoxide (CO), and oxygen (O) in the gases. Although the analysis does not directly determine the amount of nitrogen present in the flue-gases, yet its actual amount, as well as that of the air supply, may readily be ascertained by calculation. When air is drawn through an opening, like an ash-pit door, sometimes an anemometer can be used for ascertaining the velocity through the area, and the air supply be determined by these means.

A pound of carbon requires for complete combustion, 2.67 pounds of oxygen, or a volume of 32 cubic feet at 60° F. , and the gaseous product, carbon dioxide (CO_2), when cooled occupies precisely the same volume as the oxygen, viz., 32 cubic feet. If the oxygen is mixed with nitrogen in the same proportion as it is found in air (20.91 O and 79.09 N), the volume of the carbon dioxide (CO_2) after combustion, and also its proportion to nitrogen, is the same as that of the oxygen; hence, for complete combustion of carbon, with no excess of air, the volumetric analysis of the flue gases is,

Carbon dioxide	$\text{CO}_2 = 20.91\%$
Carbon monoxide	$\text{CO} = \text{None}$
Oxygen	$\text{O} = \text{None}$
Nitrogen	$\text{N} = 79.09\%$

If the supply of air is in excess of that required to supply the oxygen needed, the combined volumes of the carbon dioxide and oxygen are still the same as that of the oxygen before combustion; consequently, *for the complete combustion of pure carbon, the sum of the percentages by volume of the carbon dioxide and oxygen in the flue gases must always be 20.91, no matter what the supply of air may be.*

Carbon monoxide (CO) produced by imperfect combustion of carbon, occupies *twice* the volume of the oxygen entering into its composition, and renders the volume of the flue gases greater than that of the air supply in the proportion of

$$\frac{100}{100 - \frac{1}{2} \text{ the } \% \text{ of CO}}, \text{ hence}$$

when pure carbon is the fuel, the sum of the percentages of carbon dioxide, oxygen, and one-half the carbon monoxide must be in the same ratio to the nitrogen as is oxygen in the air, viz. 20.91 to 79.09.

Orsat Apparatus. The analysis of the flue-gases is best made for practical purposes by means of the Orsat apparatus, shown in Fig. 94. The operation is as follows: Exactly 100 cc of the gas sample are drawn into the graduated measuring burette, *A*, and then passed in succession into the U-form absorbing vessels, *D*, *E*, *F*, each time being returned to and measured in *A*. In passing into the U-shaped vessels, the gas displaces the liquid contained therein, driving it up into the other legs. A portion of the fluids, however, adheres to the glass tubes placed in the vessels for that purpose, and comes in intimate contact with the gas. Each vessel absorbs a different constituent. *D* is filled with a solution of potassium hydroxide and takes up the carbon dioxide; *E* contains pyrogallie acid, which removes the oxygen; and *F* absorbs the carbon monoxide in a solution of cuprous chloride. The reduction in volume measured in *A* gives the percentage of each constituent gas.

The connections to *A* are made through the glass stop cocks *M* and the capillary tube *C*. The movement of the gases is produced by lowering or raising the bottle *L*,

which is connected to the lower part of *A* by the rubber tube *S*, and is partially filled with water. When a measurement is taken, the level of the water in *A* and *L* must be the same, so that all measurements are taken at at-

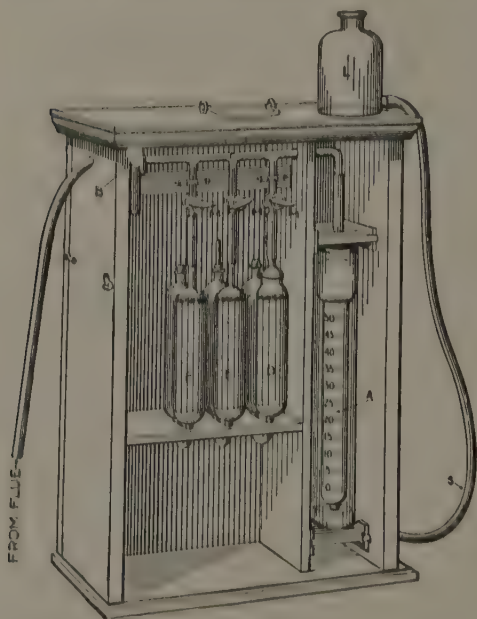


Fig. 94. Orsat Apparatus for Flue-Gas Analysis.

mospheric pressure. A constant temperature of the gas in *A* is maintained by the water in the surrounding cylinder shown.

The sample is drawn into the apparatus through the cock *B*, which also serves to connect the capillary tube

to the atmosphere, the latter connection being through the spindle of the cock, thus permits the removal of any excess of gas above 100 cc that may have been drawn into *A*. Before the sample is drawn, the vessels *D*, *E* and *F* should have their respective liquids raised to the cocks *M* (which can then be closed, and the atmospheric pressure acting through the other leg, which is open, will keep them filled); the burette *A* and the capillary tubes should be filled with water up to the cock *B*. All this can easily and quickly be done by raising and lowering *L*, and opening and closing cocks *M* and *B*. The absorption of oxygen and carbon monoxide is very slow, and the gas should be passed back and forth a number of times until a reduction of volume is no longer indicated.

As the pressure of the gases in a flue is less than the atmospheric pressure, they will not, of themselves, flow through the rubber or metal tubing connecting to the analyzing apparatus; but by filling the instrument two or three times and discharging it into the atmosphere through cock *B*, the air can be removed from the connecting tubing and a sample of the gas be obtained. For rapid work, an aspirator can be used for drawing the gas from the tube in a constant stream. If this is used there is less danger of an admixture of air. It is sometimes desirable to take a sample that represents an average during half an hour, or an hour, and in this case a metal, or glass vessel with a stop-cock at both top and bottom, and filled with water, can be connected through the upper stop-cock to the flue, and the bottom cock then be opened. The water will gradually drip out, drawing the gas into the vessel. The time taken to fill it can be regulated by the lower cock.

The result of a flue-gas analysis depends both on the manner and time of taking the sample, and to get at the average composition of the gas, a number of determinations should be made on samples from different parts of the flue.

The analysis made by the Orsat apparatus is volumetric; if the analysis by weight is required it can be found from the volumetric analysis as follows:

Multiply the percentages by volume by the molecular weight of the gas, and divide by the sum of all the products; the quotient will be the percentage by weight.

The molecular weights are as follows:

Carbon dioxide	44
Carbon monoxide	28
Oxygen	32
Nitrogen	28

CALCULATIONS FOR EFFICIENCY OF THE PLANT.

Having thus successfully conducted the test to its close, and being armed with all the data heretofore noted, the engineer is now ready to compute the results.

If the test is made for the purpose of determining the efficiency of the boiler and setting as a whole, including grate, chimney draft, etc., then the result must be based upon the number of pounds of water evaporated per pound of coal. This latter phrase includes not only the purely combustible matter in the coal, but the non-combustible also, as ash, moisture, etc. Some varieties of western coal contain as high as 12 to 14 per cent. of moisture, and the ability of the furnace to extract heat

from the mass is to be tested, as well as the ability of the boiler to absorb and transmit that heat to the water. Therefore the efficiency of the boiler and furnace =

$$\frac{\text{Heat absorbed per pound of coal.}}{\text{Heating value of one pound of coal.}}$$

EFFICIENCY OF THE BOILER.

The heating surface of the boiler must transmit heat from the hot furnace gases on one side to the water on the other. This transmission of heat is very rapid through the metal of the boiler, but the accumulation of scale on the interior and soot on the exterior surfaces greatly obstructs the flow of heat and renders the heating surface inefficient.

If the test is to determine the efficiency of the boiler itself as an absorber of heat, then the combustible alone must be considered in working out the final result. Thus,

Efficiency of boiler =

$$\frac{\text{Heat absorbed per pound of combustible.}}{\text{Heating value of one pound of combustible.}}$$

When making a series of tests for the purpose of comparing the economical value of different varieties of coal the conditions should be as nearly uniform as possible; that is, let the tests be made under ordinary working conditions and with the same boiler or boilers, and if possible with the same fireman.

The following is a record of one of many evaporation tests made by the author, and is introduced here for the purpose of illustrating methods of computing the results to be obtained from the various data. The rather

large quantity of coal burned per square foot of grate surface per hour (25 lbs.) is owing to the fact that the boiler was run to its full capacity, the coal burning clean and forming no clinker. The chimney draft also was exceptionally good, giving a large unit of evaporation per square foot of heating surface per hour. The low temperature of the escaping gases is due to the fact that they were returned over the top of the boiler before passing to the chimney.

Date of test.....

Duration of test, 12 hours.

Boiler, return tubular, 72 in. diameter, 18 ft. long,
62-4½ in. tubes.

Kind of coal, Pocahontas ; average steam		
pressure	85	lbs.
Weight of coal consumed	11,100	"
Weight of water apparently evaporated.....	107,187	"
Weight of dry ash returned.8.1 per cent =	900	"
Moisture in the coal.....2.0 " =	222	"
Moisture in the steam....1.0 " =	1,071	"
Dry coal corrected for moisture.....	10,878	"
Weight of combustible.....	9,978	"
Water corrected for moisture in the steam.....	106,116	"
Water evaporated into dry steam, from and at 212°	117,788	"
Water evaporated per lb. of coal, actual conditions	9.65	"
Water evaporated per lb. of coal, from and at 212°.....	10.61	"
Water evaporated per lb. of combustible, from and at 212°.....	11.81	"

Water evaporated per lb. of dry coal, from and at 212°	10.82 lbs.
Water evaporated per hr. per sq. ft. of heating surface	6.22 "
Coal burned per sq. ft. of grate surface per hour	25 "
Horse power developed by boiler during test	284.5
Temperature of feed water, average....	141°
Temperature of chimney gases, average..	400°
Square feet of grate surface	36
Square feet of heating surface	1,576
Ratio of grate surface to heating surface	43.7

The results obtained will be taken up in their regular order beginning with, first, water evaporated into dry steam from and at 212°. As it may be of benefit to some, a short definition of the meaning of the above expression is here given.

The term "equivalent evaporation," or the evaporation from and at 212°, assumes that the feed water enters the boiler at a temperature of 212° and is evaporated into steam at 212° temperature and at atmospheric pressure. As for instance, if the top man hole plate were left out or some other large opening in the steam space allowed the steam to escape into the atmosphere as fast as it was generated. Owing to the variation in the temperatures of the feed water used in different tests, and also the variation in the steam pressure, it is absolutely necessary that the results of all tests be brought by computation to the common basis of 212° in order to obtain a just comparison.

The process by which this is done is as follows: Referring to the record of the test it is seen that the steam pressure average was 85 lbs. gauge pressure, or 100 lbs. absolute, and that the temperature of the feed water was 141°. Referring again to Table 10, physical properties of steam, it will be seen that in a pound of steam at 100 lbs. absolute pressure there are 1,181.8 heat units, and in a pound of water at 141° temperature there are 109.9 heat units. It therefore took $1,181.8 - 109.9 = 1,071.9$ heat units to convert one pound of feed water at 141° into steam at 85 lbs. pressure. To convert a pound of water at 212° into steam at atmospheric pressure, and 212° temperature requires 965.7 heat units, and the 1,071.9 heat units would evaporate $1,071.9 \div 965.7 = 1.11$ lbs. water from and at 212°. The 1.11 is the factor of evaporation for 85 lbs. gauge pressure and 141° temperature of feed water, and by multiplying "water corrected for moisture in the steam" (see record of test), 106,116 lbs., by 1.11, the weight of water which could have been evaporated into steam from and at 212° is obtained, which is 117,788 lbs. The factor of evaporation is based upon the steam pressure and the temperature of the feed water in any test and the formula for ascertaining it is as follows: Factor = $\frac{H - h}{965.7}$, in which

H = total heat in the steam and h = total heat in the feed water. It is used in shortening the process of finding the evaporation from and at 212°, and Table 21 gives the factor of evaporation for various pressures and temperatures.

Second, water evaporated per pound of coal actual conditions = water apparently evaporated divided by coal consumed = 9.65 lbs. No accurate estimate regard-

TABLE 21.

FACTORS OF EVAPORATION.

Feed Water Temperature.	Gauge Press. 50 lbs.	Gauge Press. 60 lbs.	Gauge Press. 70 lbs.	Gauge Press. 80 lbs.	Gauge Press. 90 lbs.	Gauge Press. 100 lbs.	Gauge Press. 110 lbs.	Gauge Press. 120 lbs.	Gauge Press. 140 lbs.
212°	1.027	1.030	1.032	1.035	1.037	1.039	1.041	1.043	1.047
200°	1.039	1.042	1.045	1.047	1.050	1.052	1.054	1.056	1.059
191°	1.049	1.052	1.054	1.057	1.059	1.061	1.063	1.065	1.069
182°	1.058	1.061	1.064	1.066	1.069	1.071	1.073	1.075	1.078
173°	1.067	1.070	1.073	1.076	1.078	1.080	1.082	1.084	1.087
164°	1.077	1.080	1.083	1.085	1.087	1.090	1.091	1.093	1.097
152°	1.089	1.092	1.095	1.098	1.100	1.102	1.104	1.106	1.109
143°	1.099	1.102	1.105	1.107	1.109	1.111	1.113	1.115	1.119
134°	1.108	1.111	1.114	1.116	1.119	1.121	1.123	1.125	1.128
125°	1.118	1.121	1.123	1.126	1.128	1.130	1.132	1.134	1.137
113°	1.130	1.133	1.136	1.138	1.140	1.143	1.145	1.146	1.150
104°	1.138	1.142	1.145	1.148	1.150	1.152	1.154	1.156	1.159
95°	1.149	1.152	1.154	1.157	1.159	1.161	1.163	1.165	1.169
86°	1.158	1.161	1.164	1.166	1.169	1.171	1.173	1.174	1.178
77°	1.167	1.170	1.173	1.176	1.178	1.180	1.182	1.184	1.187
65°	1.180	1.183	1.186	1.188	1.190	1.192	1.194	1.196	1.200
56°	1.189	1.192	1.195	1.197	1.200	1.202	1.204	1.206	1.209
47°	1.199	1.201	1.204	1.207	1.209	1.211	1.213	1.215	1.218
38°	1.208	1.211	1.214	1.216	1.218	1.220	1.222	1.224	1.228

ing the quality of the coal or the efficiency of the boiler can be made from this figure (9.65 lbs.). It can be used, however, in estimating the cost of fuel for generating the steam; as, for instance, if the boiler is supplying steam to an engine that uses 30 lbs. of steam per horse power per hour, it will require $30 \div 9.65 = 3.1$ lbs. of coal per horse power per hour; the "actual conditions" under which the boiler is being operated being the pres-

sure of steam required by the engine and the temperature of the feed water.

Third, water evaporated per pound of coal from and at 212° = water evaporated into dry steam from and at $212^{\circ} \div$ coal consumed = 10.61 lbs. This figure is the proper one to use in comparing the relative economic values of different varieties of coal tested with the same boiler or boilers.

Fourth, water evaporated per pound of combustible from and at 212° = water evaporated into dry steam from and at $212^{\circ} \div$ weight of combustible = 11.81 lbs. This result is the one to be used for ascertaining the efficiency of the boiler, and the percentage of efficiency is found by dividing the heat absorbed by the boiler per pound of combustible by the heat value of one pound of combustible. The average heat value of bituminous and semi-bituminous coals is not far from 15,000 heat units per pound of combustible. In the evaporation of 11.81 pounds of water from and at 212° the heat absorbed was $11.81 \times 965.7 = 11,404.9$ heat units. The efficiency of the boiler therefore was $\frac{11,404.9 \times 100}{15,000}$ = 76 per cent.

In like manner to ascertain the efficiency of the boiler and furnace as a whole, the water evaporated from and at 212° per pound of coal is taken. Thus, $10.61 \times 965.7 = 10,246$ heat units absorbed from each pound of coal. Now assuming that there were 13,500 heat units in each pound of the coal used in the test, the per cent of efficiency of boiler and furnace was

$$\frac{10,246 \times 100}{13,500} = 75.9.$$

Fifth, water evaporated per pound of dry coal from and at 212° : water evaporated into dry steam from and at 212° divided by coal corrected for moisture. Thus, $117,788 : 10.878 = 10.82$ lbs. This result is useful for calculating the results of tests of the same grade of coal, but differing in the degree of moisture in each.

Sixth. Boiler horse power. The latest decision of the American Society of Mechanical Engineers (than whom there is no better authority) regarding the horse power of a boiler is as follows: "The unit of commercial horse power developed by a boiler shall be taken as $34\frac{1}{2}$ units of evaporation per hour. That is, $34\frac{1}{2}$ lbs. of water evaporated per hour from a feed temperature of 212° into steam of the same temperature. This standard is equivalent to 33,317 B. T. U. per hour. It is also, practically equivalent to an evaporation of 30 lbs. of water from a feed water temperature of 100° F. into steam of 70 lbs. gauge pressure."

According to this rule the horse power developed by the boiler during the test under consideration: - water evaporated into dry steam from and at 212° , 117,788 lbs. $\div$ 12 hrs. $\div$ $34.5 = 284.5$ horse power.

QUESTIONS.

78. Q. What is one of the most important duties of the engineer when he goes on watch?

A. To ascertain the exact height of the water in his boilers.

79. Q. Describe the correct method of doing this.

A. Open the valve in the drain pipe of the water column, and allow the water to blow out freely for a few seconds, then close the valve and note the level of the water when it settles back in the gauge glass.

80. Q. What is the next important step in beginning the day's work?

A. To see that the fires are cleaned and in good condition.

81. Q. In firing boilers by hand, what is the first and most important rule to be observed?

A. Keep a clean fire.

82. Q. What is the second rule?

A. See that every square inch of grate surface is covered with a good live fire.

83. Q. Give the third rule regarding firing by hand.

A. Keep a level fire.

84. Q. What is the fourth rule?

A. When cleaning the fire, always clean all clinkers and dead ashes away from the back end of the grates and the bridge wall.

85. Q. Why should this be done?

A. In order to allow a free passage of the air through the grate bars, so as to promote combustion.

86. Q. What is one of the greatest aids to good combustion in a hand-fired furnace?

A. A clean bridge wall, kept as hot as possible.

87. Q. What precautions should be observed regarding the depth of the fire?

A. It should not be allowed to become so deep and heavy as to prevent the air from passing up through it freely.

88. Q. How should the position of the ash-pit doors be regulated?

A. With a clean, light fire, a slight opening will be sufficient, but with a heavy fire, and the grates clogged with ashes, a larger opening is necessary.

89. Q. How can the best results be secured in firing bituminous coal?

A. By leaving the fire doors slightly open for a few seconds immediately after throwing in a fire.

90. Q. What reason is there for doing this?

A. Because the volatile matter in the coal flashes into flame the instant it comes in contact with the heat of the furnace, and unless there is a sufficient supply of oxygen present just then, the combustion will be imperfect.

91. Q. What is the result of this imperfect combustion?

A. The formation of carbonic oxide gas, and the consequent loss of about two-thirds of the heat units contained in the coal.

92. Q. How may this loss be prevented in a great measure?

A. By admitting a sufficient volume of air, either through the fire doors, directly after throwing in a fresh fire, or, better still, providing air ducts through the bridge

wall, or side walls, which will direct the air in on top of the fire.

93. Q. How much air is required for the complete combustion of one pound of coal?

A. By weight 12 pounds—by volume 150 cubic feet.

94. Q. What precaution is necessary regarding the tubes of a boiler in order to get the best results from the fuel?

A. The tubes should be kept clean and free from soot and scale.

95. Q. Should the steam jet cleaner be depended upon alone for cleaning the tubes?

A. No. The scraper should also be used.

96. Q. How should safety valves be looked after?

A. They should be ample in size, never overloaded, and should be tested at least once a day to see that they act freely.

97. Q. At what point should the steam gauge pointer stand when the pressure is off?

A. It should stand at zero.

98. Q. What should be done in case of low water in a boiler?

A. The fire should be covered immediately with ashes, earth, or if neither is available use fresh coal. Draw the fire as soon as it can be done without increasing the heat.

99. Q. Should the rate of feeding the water be increased, in case of extremely low water in the boiler?

A. It should not, neither should the engine be stopped or the safety valve lifted, until the fires are out, and the boiler cooled down.

100. Q. In case of indications of cracks or blisters appearing on the boiler sheets, what should be done?

A. There should be no delay in making repairs.

101. Q. What should be done with fusible plugs when used?

A. They should be cleaned and carefully scraped on both water and fire sides at each washing out.

102. Q. How may the most economical results regarding fuel be attained with a steam boiler?

A. By keeping the heating surfaces clean, both inside and outside, also careful firing, a little at a time, but keeping the grates covered.

103. Q. Should cold water ever be fed into a boiler when it is under pressure?

A. Not when it can be avoided.

104. Q. How may foaming usually be stopped?

A. By checking the outflow of steam, by blowing down and pumping up, or by checking the draft and fires.

105. Q. Should air be allowed to pass to the boiler or tubes, except through the furnace?

A. It should not, as it will cause a waste of fuel.

106. Q. What should be done with leaks when discovered?

A. They should be repaired as soon as possible.

107. Q. What precautions should be observed when preparing to empty a boiler for washing out or other purposes?

A. Allow it to cool down until there is no steam pressure, and until the brick work is cool also.

108. Q. When firing up a boiler what course should be pursued?

A. Steam should be raised very slowly and rapid firing avoided.

109. Q. What bad results follow too rapid firing up of a boiler?

A. Straining of the joints and seams caused by unequal expansion.

110. Q. What should be done with a boiler that is to stand idle for any length of time?

A. It should be emptied and thoroughly dried. In case this is impracticable, fill it full of water, and put in a quantity of washing soda.

111. Q. How long a time may a boiler be safely operated between dates of washing out?

A. This depends upon the nature of the feed water. The time should never be longer than two weeks, and with very bad water, the boiler should be washed out once a week.

112. Q. Besides cleaning the boiler inside, what other very important work should the boiler washer perform while inside the boiler?

A. He should closely examine all braces, stays, and rivets by tapping them with a hammer. Any loose or defective parts can usually be detected in this way.

113. Q. Describe four ways in which tube failures may occur?

A. 1. Pitting. 2. Defective welds. 3. Bagging. 4. Scabbing and blistering.

114. Q. How may a great saving in fuel be effected with regard to the feed water?

A. By heating it with the exhaust steam from engines and pumps before passing it to the boilers.

115. Q. How many kinds of exhaust heaters are there?

A. Two. Open and closed.

116. Q. What principle governs the operation of an open heater?

A. The exhaust steam mingles with the water, and heats it as it passes through the heater.

117. Q. Describe a closed heater?

A. In a closed heater the steam, either exhaust or live steam, passes through tubes inside a shell containing the feed water. In some types of closed heaters the water passes through the tubes and the steam is passed through the shell.

118. Q. What should be the capacity of a closed heater relative to the demands of the boiler for water?

A. It should contain water sufficient to supply the boiler for fifteen or twenty minutes.

119. Q. Describe the available heating surface of a stationary boiler, of either type, return tubular or water tube?

A. The lower half of the shell, and heads, and the combined cross sectional area of all the tubes.

120. Q. What should be the location of the water gauge glass relative to the water level in the boiler?

A. It should be located at such a height as to bring the lower end of the glass tube on a level with the danger point for low water in the boiler.

121. Q. Where should the lower gauge cock be located relative to the danger point?

A. About three inches above.

122. Q. Should an engineer or water tender depend entirely upon the water gauge glasses?

A. He should not, but should frequently open and try the gauge cocks.

123. Q. What should be done with the entire water column several times a day?

A. It should be blown out thoroughly.

124. Q. What should be done with the safety valves in order to make them reliable?

A. They should be allowed to blow off at least twice a week.

125. Q. Why is this necessary?

A. Because the valves are liable to become corroded and stick to their seats if not attended to properly.

126. Q. What is the rule for finding the bursting pressure of boilers?

A. Multiply the tensile strength by the thickness and divide by one-half the diameter of the shell.

127. Q. How may the safe working pressure of a boiler be ascertained?

A. By dividing the bursting pressure by five.

128. Q. What is the rule for ascertaining the velocity of flow in a pump?

A. Multiply the number of strokes per minute by length of stroke in feet. This will give piston speed.

129. Q. How may velocity of flow in the discharge pipe of a pump be found?

A. Divide square of diameter of water piston by the square of the diameter of pipe, and multiply by piston speed per minute.

130. Q. What is the rule for finding velocity in feet per minute required to discharge a given quantity of water in a given time?

A. Multiply number of cubic feet to be discharged by 144 and divide by area of pipe in inches.

131. Q. When the volume and velocity of water to be discharged are known, how may the area of the pipe be ascertained?

A. Multiply volume in cubic feet by 144 and divide by velocity in feet per minute.

132. Q. How many different classes of mechanical stokers are there?

A. Four, viz., the endless chain grate. Second, those in which the grates have an up and down motion. Third, inclined grates, and fourth, under feed stokers.

133. Q. For what two purposes are chimneys designed?

A. First, to carry off obnoxious gases. Second, to produce a draft and so facilitate combustion.

134. Q. How may the first purpose be served?

A. By size.

135. Q. How is the second requisite met?

A. By height.

135a. Q. What is the best form of chimney?

A. Round and straight or slightly tapering.

136. Q. Of what is air composed?

A. Air is a mixture of the two gases, oxygen and nitrogen.

137. Q. In what proportions by volume?

A. 20.91 parts oxygen to 79.09 parts nitrogen.

138. Q. In what proportions by weight?

A. 23.15 parts oxygen to 76.85 parts nitrogen.

139. Q. What is meant by the term combustion as used in steam engineering?

A. It is the rapid chemical combination of oxygen with the carbon, hydrogen and sulphur in the fuel with the accompaniment of heat and light.

140. Q. What is meant by the symbol CO_2 ?

A. CO_2 represents perfect combustion, viz., the creation of carbon dioxide.

141. Q. What is the most abundant combustible in nature?

A. Carbon.

142. Q. How many heat units are contained in one pound of pure carbon?

A. 14,500.

143. Q. What is the heating value of one pound of hydrogen gas?

A. 62,000.

144. Q. Give the composition of coal.

A. Fixed carbon, volatile matter, ash and sulphur in various proportions, depending upon the quality of the coal.

145. Q. Is sulphur desirable as a constituent of coal?

A. It is not. The gases formed from its combustion attack the metal of the boiler, causing corrosion.

146. Q. What office does nitrogen perform in combustion?

A. No useful office. Rather it is a detriment, and in fact is the chief source of loss in furnaces. It is drawn in with the air.

147. Q. What is meant by the term calorific value of fuel?

A. The amount of heat liberated per pound of fuel undergoing perfect combustion?

148. Q. What are economizers in connection with a boiler plant?

A. Coils or stacks of cast iron pipe placed within the smoke flue or breeching and surrounded by the hot gases while the water is passed through the pipes on its way to the boilers, thus receiving an additional amount of heat.

149. Q. What two factors are necessary in order to attain economy in the burning of coal?

A. A constant high furnace temperature and quick combustion.

150. Q. Define the term heat?

A. Heat is the result of the vibration of the molecules or atoms composing matter.

151. Q. Upon what does the intensity of heat depend?

A. Upon the rapidity of the agitation to which the molecules are subject.

152. Q. What are the general effects of heat?

A. When heat is added to or taken away from a body the temperature of the body is altered and its volume is varied.

153. Q. What is absolute zero?

A. It is that degree of temperature at which, owing to the intense cold, a perfect gas would disappear. Absolute zero is 461° below the zero of the Fahrenheit thermometer.

154. Q. What is a heat unit (B. T. U.)?

A. It is the quantity of heat required to raise the temperature of one pound of water one degree, or from 39° to 40° F.

155. Q. What is the mechanical equivalent of heat?

A. 778 foot pounds; in other words, 778 pounds raised one foot.

156. Q. What is specific heat of any substance?

A. The ratio of the quantity of heat required to raise a given weight of that substance one degree in temperature, to the quantity of heat required to raise an equal weight of water one degree when the water is at its maximum density, viz., 39.1° F.

157. Q. What is latent heat?

A. Heat given to a body and not warming it.

158. Q. What is sensible heat?

A. Heat given to a body and warming it.

159. Q. Of what is pure water composed?

A. By volume—Hydrogen 2 parts, oxygen 1.

By weight— “ 11.1 “ “ 88.9.

160. Q. Is perfectly pure water desirable for use in a steam boiler?

A. It is not, as it will cause corrosion and pitting of the sheets.

161. Q. What two ingredients in water are the chief causes of incrustation in boilers?

A. The carbonates of lime and magnesia.

162. Q. What is steam?

A. Steam is the vapor of water generated by an increase of the natural vibrations of molecules of the water through the application of heat.

163. Q. What is saturated steam?

A. Steam taken directly from the boiler to the engine without being superheated.

164. Q. What is superheated steam?

A. Steam that has been heated to a higher temperature than that due to its pressure.

165. Q. What should be done with all pipes through which live steam is conducted for purposes of heating or power?

A. They should be well protected by a covering to prevent condensation.

166. Q. What is the object of an evaporation test?

A. To ascertain how many pounds of water the boilers are evaporating per pound of fuel burned.

167. Q. What is a calorimeter?

A. An apparatus for determining the amount of moisture there is in steam.

168. Q. How is the efficiency of the boiler and furnace ascertained through an evaporation test?

A. By weighing the coal consumed and the water

evaporated during a certain number of hours and dividing the number of lbs. of water evaporated by the number of lbs. of coal consumed. This will give number of lbs. water evaporated per lb. of coal.

169. Q. What is meant by boiler horse power?

A. The evaporation of $34\frac{1}{2}$ lbs. water from a feed temperature of 212° into steam of the same temperature, or the evaporation of 30 lbs. water from a feed temperature of 100° into steam at 70 lbs. gauge pressure.

STEAM ENGINES.

DESCRIPTIVE.

Steam engines may be divided into two general classes, viz., simple and compound.

A simple engine may be either condensing or non-condensing, but its leading characteristic is, that the steam is used in but one cylinder, and from thence it is exhausted either into the atmosphere or into a condenser.

A compound engine is one in which the steam is made to do work in two or more cylinders before it is allowed to exhaust, and this class of engine may be either condensing or non-condensing.

In a non-condensing engine the pressure of the atmosphere, amounting to 14.7 lbs. per square inch at sea level, is constantly in resistance to the motion of the piston. Therefore the exhaust pressure cannot fall below the atmospheric pressure, and is generally from two to five pounds above it, caused by the resistance of bends and turns in the exhaust pipe, or other causes which tend to retard the free passage of the steam.

The advantage, from an economical point of view, of exhausting the steam into a condenser in which a vacuum is maintained, will be fully set forth in the section on Indicator Work.

Condensers are of two classes, viz., jet condensers and surface condensers.

In a jet condenser the steam is exhausted into an air-tight iron vessel of any convenient shape, generally cylindrical and of suitable size, and is there condensed by coming in contact with a jet of cold water, admitted

in the form of a spray. The air pump, which also maintains a vacuum in the condenser, draws this water, together with the condensed steam, away from the condenser.

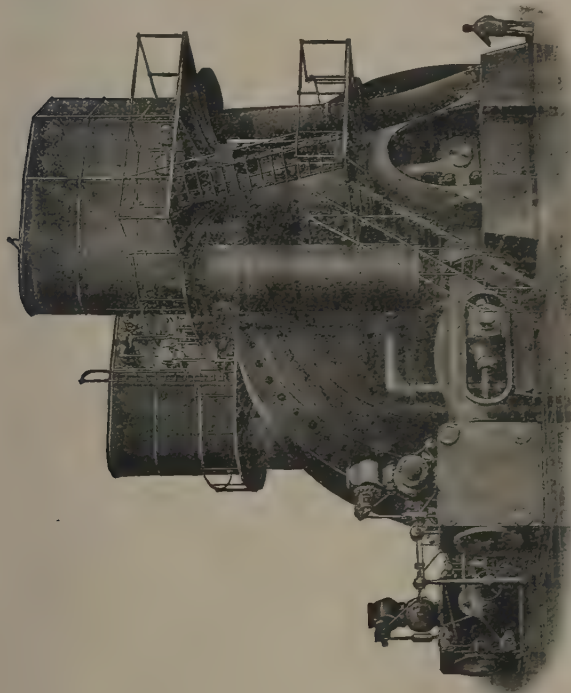


Fig. 95. Reynolds Combined Vertical and Horizontal Engine 12,000 Horse Power Cylinders, 44x88x60. Built by Allis-Chalmers Company.

The surface condenser, like the jet condenser, consists of an air-tight iron vessel, either cylindrical or rectangular in shape, but unlike the jet condenser, it is fitted with a large number of brass or copper tubes of

small diameter, through which cold water is forced by a pump, called a circulating pump. A vacuum is also maintained in the body of the condenser by the air pump, and the steam exhausting into this is condensed by coming in contact with the cool surface of the tubes. Or, as is often the case, the exhaust steam passes through the tubes in place of around them, and the condensing water is forced into and through the body of the condenser, the vacuum in this case being maintained in the tubes. Owing to the fact that in a surface condenser the steam does not mix with the water, a larger quantity of condensing water is required than in a jet condenser, but on the other hand, an advantage is gained by having the pure water of condensation; in other words, the condensed steam, which may be returned to the boilers along with the regular feed water supply, and will greatly aid in preventing the formation of scale, while the water of condensation as it comes from a jet condenser, being mixed with oil and other impurities, is not, as a rule, suitable to be fed to boilers.

There are many different types of jet condensing apparatus, in some of which no air pump is used; their action being based somewhat upon the principle of the injector used for feeding boilers. In this type of jet condenser the supply of condensing water is drawn from outside pressure, either from an overhead tank or other source, and passing into an annular enlargement of the exhaust pipe, is discharged downwards in the form of a cylindrical sheet of water into a nozzle which gradually contracts. The exhaust steam, entering at the same time, is condensed, and the contracting neck of the cone shaped nozzle gradually brings the water to a solid jet, and it rushes through the nozzle with a velocity

sufficient to create a vacuum. This type of condenser can only be used where the discharge pipe has a free outlet.

The jet condenser with air pump attached is the most reliable as well as economical for general purposes, for the reason that with this type the supply of condensing water may be drawn from a well or other source lower than the level of the condenser. These condensers are also generally fitted with a "force injection," as it is called, which is simply a connection between the condenser and water main or tank, for the purpose of letting cold water into the condenser to condense the exhaust steam when starting the engine, and thus aid in forming a vacuum. When a good vacuum has been established and the engine is running up to speed, the force injection may be shut off, and the water will flow into the condenser from the well by suction. The above refers to engines in which the air pump receives its motion directly from the engine.

Another type of jet condensing apparatus is the independent air pump and condenser, which is still better, for the reason that the air pump, which is simply an ordinary double acting steam pump, may be started independently of the engine, and, in fact, before the engine is started, thus creating a vacuum in the condenser, and greatly facilitating the starting of the engine. Another great advantage in the independent condensing apparatus is, that there is not so much danger of the water backing up into the cylinder in case of a sudden shut down of the engine, because the air pump may be kept in operation, thus relieving the condenser of water; whereas, if the air pump gets its motion from the engine, it will of course stop when the engine stops, and unless

the injection water is shut off immediately after closing the throttle there is great danger of the cylinder becoming flooded with water, resulting very often in a broken cylinder head, or a bent piston rod.

The quantity of water required to condense the exhaust steam of an engine is determined by three factors: First, the density, temperature and volume of the steam to be condensed in a given time: second, the temperature of the overflow or discharge, and third, the temperature of the injection water. For instance, the temperature

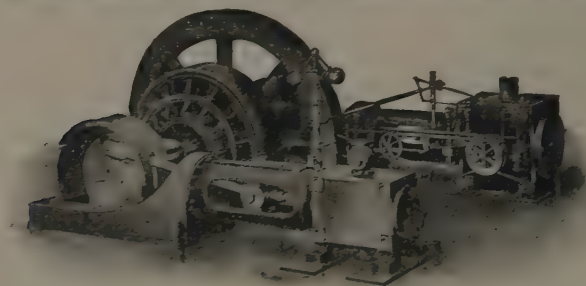


Fig. 96. Cross Compound Direct Connected Corliss Engine, Allis-Chalmers Co.

of the injection water may be 35° in the winter and 70° in the summer. Or it may be desired to keep the overflow at as high a temperature as possible for the purpose of feeding the boilers. Again, the pressure, and consequently the temperature of the exhaust steam as it enters the condenser, varies with different engines, and often with the same engine, according as the load is light or heavy. Therefore the only accurate method of estimating the amount of condensing water required per minute or per hour, under any and all conditions, is to first

ascertain the weight of water required to condense one pound weight of steam at the temperature and pressure at which the steam is being exhausted. In these calculations the total heat in the steam must be considered. This means not only the sensible heat, but the latent heat also.

The formula for solving the above problem may be expressed as follows: $\frac{H-T}{T-I}=W$, in which

H=total heat in the steam,

T=temperature of the overflow,

I=temperature of the injection water,

W=weight of water required to condense one pound weight of steam.

To illustrate, suppose the absolute pressure of the exhaust, as shown by the indicator diagram, is 7 lbs. Referring to Table 10, it will be seen that the total heat in steam at 7 lbs. absolute is 1135.9 heat units. Assume the temperature of the overflow to be 110°, which is as high as is consistent with a good vacuum. Now the total heat to be absorbed from each pound weight of steam in this case would be 1135.9—110 = 1025.9 B. T. U.

Suppose the temperature of the condensing water to be 55° and the temperature of the overflow being 110°, there will be 110°—55° = 55° of heat absorbed by each pound of water passing into and through the condenser, and the number of pounds of water required to condense one pound weight of steam under the above conditions will equal the number of times 55 is contained in 1025.9. Expressed in plain figures the calculation is

$$\frac{1135.9-110}{110-55} = 18.65 \text{ lbs.}$$

In order to ascertain the quantity of condensing water

required per horse power per hour, it is only necessary to know the number of pounds weight of steam consumed by the engine per horse power per hour, as shown by the indicator diagram, and multiply this by the weight of condensing water required per pound of steam, as found by the above solution.

Thus, suppose the steam consumption of the engine to be 17 lbs. per I. H. P. per hour. Then $17 \div 18.65 = 317.65$ lbs. per hour, which reduced to gallons = 38.2 gals.

Or, if the steam consumption is not known, and the weight only of condensing water required per hour is



Fig. 97. Tandem Compound Engine, Buckeye Engine Co.

desired, regardless of the horse power developed by the engine, it will be necessary, first, to estimate the total volume of steam exhausted per hour and calculate its weight from its known pressure.

Thus, assume the engine to be 24 $\times$ 48 in., and the R. P. M. to be 80. Then the piston displacement will equal area of piston less one-half area of rod multiplied by length of stroke. Referring to Table 22, the area of a circle 24 in. in diameter = 452.39 sq. in. Suppose the piston rod to be 4.5 in. in diameter, its area, according to Table 22, is 15.904 sq. in., one-half of which = 7.952

sq. in. The effective area of the piston now becomes $452.39 - 7.952 = 444.43$ sq. in., and the piston displacement equals $444.43 \times 48 = 21332.64$ cu. in. Dividing this

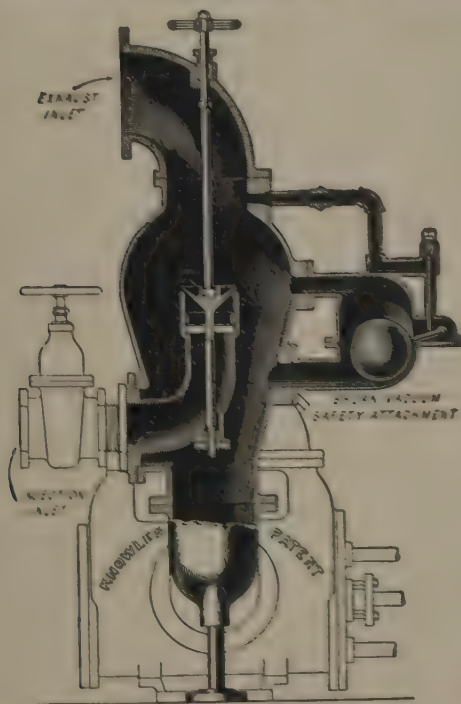


Fig. 98. Knowles Jet Condenser.

by 1728 (number of cubic inches in a cubic foot) gives 12.34 cu. ft. of piston displacement. The total volume of steam exhausted per minute, therefore, will be $12.34 \times 2 \times 80 = 1974.4$ cu. ft.

The absolute pressure of the exhaust may again be

assumed to be 7 lbs. per square inch. Referring to Table 10, the weight of one cubic foot of steam at 7 lbs. absolute is .0189 lbs., and the total weight of steam exhausted per minute, therefore, would be $1974.4 \times .0189 = 37.3$ lbs., and if 18.65 lbs. water is required to condense one pound of steam, the quantity required per minute would be $37.3 \times 18.65 = 695.8$ lbs., or per hour, 41748 lbs., equal to 5029 gals. This is at the rate of 8.7 lbs., or a little more than one gallon per revolution for a 24×48 in. simple condensing engine. Table 23 gives the quantity of injection water required per revolution for different types of condensing engines.

TABLE 22. AREAS AND CIRCUMFERENCES OF CIRCLES.

Diam.	Area.	Circum.	Diam.	Area.	Circum.
.25	.049	.7854	8	50.265	25.132
.5	.1963	1.5708	8.25	53.456	25.918
1.0	.7854	3.1416	8.5	56.745	26.703
1.25	1.2271	3.9270	9	63.617	28.274
1.5	1.7671	4.7124	9.25	67.200	29.059
2	3.1416	6.2832	9.5	70.882	29.845
2.25	3.9760	7.0686	10	78.540	31.416
2.5	4.9087	7.8540	10.25	82.516	32.201
3	7.0686	9.4248	10.5	86.590	32.986
3.25	8.2957	10.210	11	95.033	34.557
3.5	9.6211	10.995	11.25	99.402	35.343
4	12.566	12.566	11.5	103.869	36.128
4.25	14.186	13.351	12	113.097	37.699
4.5	15.904	14.137	12.25	117.859	38.484
5	19.635	15.708	12.5	122.718	39.270
5.25	21.647	16.493	13	132.732	40.840
5.5	23.758	17.278	13.25	137.886	41.626
6	28.274	18.849	13.5	143.130	42.411
6.25	30.679	19.635	14	153.938	43.982
6.5	33.183	20.420	14.25	159.485	44.767
7	38.484	21.991	14.5	165.130	45.553
7.25	41.282	22.776	15	176.715	47.124
7.5	44.178	23.562	15.25	182.654	47.909

TABLE 22.—Continued.

Diam.	Area.	Circum.	Diam.	Area.	Circum.
15.5	188.692	48.694	28.5	637.941	89.535
16	201.062	50.265	29	660.521	91.106
16.25	207.394	51.051	29.25	671.958	91.891
16.5	213.825	51.836	29.5	683.494	92.677
17	226.980	53.407	30	706.860	94.248
17.25	233.705	54.192	30.25	718.690	95.033
17.5	240.520	54.978	30.5	730.618	95.818
18	254.469	56.548	31	754.769	97.389
18.25	261.587	57.334	31.25	766.992	98.175
18.5	268.803	58.119	31.5	799.313	98.968
19	283.529	59.690	32	804.249	100.53
19.25	291.039	60.475	32.25	816.86	101.31
19.5	298.648	61.261	33	855.30	103.67
20	314.160	62.832	33.25	868.30	104.45
20.25	322.063	63.617	33.5	881.41	105.24
20.5	330.064	64.402	34	907.92	106.81
21	346.361	65.973	34.25	921.32	107.60
21.25	354.657	66.759	34.5	934.82	108.38
21.5	363.051	67.544	35	962.11	109.95
22	380.133	69.115	35.25	975.90	110.74
22.25	388.822	69.900	35.5	989.80	111.52
22.5	397.608	70.686	36	1017.8	113.09
23	415.476	72.256	36.25	1032.06	113.88
23.25	424.557	73.042	36.5	1046.35	114.66
23.5	433.731	73.827	37	1075.21	116.23
24	452.390	75.398	37.25	1089.79	117.01
24.25	461.864	76.183	37.5	1104.46	117.81
24.5	471.436	76.969	38	1134.11	119.38
25	490.875	78.540	38.25	1149.08	120.16
25.25	500.741	79.325	38.5	1164.15	120.95
25.5	510.706	80.110	39	1194.59	122.52
26	530.930	81.681	39.25	1209.95	123.30
26.25	541.189	82.467	39.5	1225.42	124.09
26.5	551.547	83.252	40	1256.64	125.66
27	572.556	84.823	40.25	1272.39	126.44
27.25	583.208	85.608	40.5	1288.25	127.23
27.5	593.958	86.394	41	1320.25	128.80
28	615.753	87.964	41.25	1336.40	129.59
28.25	626.798	88.750	41.5	1352.65	130.37

TABLE 22.—Continued.

Diam.	Area.	Circum.	Diam.	Area.	Circum.
42	1385.44	131.94	55	2375.83	172.78
42.25	1401.98	132.73	55.25	2397.48	173.57
42.5	1418.62	133.51	55.5	2419.22	174.35
43	1452.20	135.08	56	2463.01	175.92
43.25	1469.13	135.87	56.25	2485.05	176.71
43.5	1486.17	136.65	56.5	2507.19	177.5
44	1520.53	138.23	57	2551.76	179.07
44.25	1537.86	139.01	57.25	2574.19	179.85
44.5	1555.28	139.80	57.5	2596.72	180.64
45	1590.43	141.37	58	2642.08	182.21
45.25	1608.15	142.15	58.25	2664.91	182.99
45.5	1625.97	142.94	58.5	2687.83	183.78
46	1661.90	144.51	59	2733.97	185.35
46.25	1680.01	145.29	59.25	2757.19	186.14
46.5	1698.23	146.08	59.5	2780.51	186.92
47	1734.94	147.65	60	2827.44	188.49
47.25	1753.45	148.44	60.25	2851.05	189.28
47.5	1772.05	149.22	60.5	2874.76	190.06
48	1809.56	150.79	61	2922.47	191.64
48.25	1828.46	151.58	61.25	2946.47	192.42
48.5	1847.45	152.36	61.5	2970.57	193.21
49	1885.74	153.93	62	3019.07	194.78
49.25	1905.03	154.72	62.25	3043.47	195.56
49.5	1924.42	155.50	62.5	3067.96	196.35
50	1963.50	157.08	63	3117.25	197.92
50.25	1983.18	157.86	63.25	3142.04	198.71
50.5	2002.96	158.65	63.5	3166.92	199.50
51	2042.82	160.22	64	3216.99	201.06
51.25	2062.90	161.00	64.25	3242.17	201.85
51.5	2083.07	161.79	64.5	3267.46	202.68
52	2123.72	163.36	65	3318.31	204.20
52.25	2144.19	164.14	65.25	3343.88	204.99
52.5	2164.75	164.19	65.5	3369.56	205.77
53	2206.18	166.50	66	3421.20	207.34
53.25	2227.05	167.29	66.25	3447.16	208.13
53.5	2248.01	168.07	66.5	3473.23	208.91
54	2290.22	169.64	67	3525.66	210.49
54.25	2311.48	170.43	67.25	3552.01	211.27
54.5	2332.83	171.21	67.5	3578.47	212.06

TABLE 22.—Continued.

Diam.	Area.	Circum.	Diam.	Area.	Circum.
68	3631.68	213.63	81.5	5216.82	256.04
68.25	3658.44	214.41	82	5281.02	257.61
68.5	3685.29	215.20	82.5	5345.62	259.18
69	3739.28	216.77	83	5410.62	260.75
69.25	3766.43	217.55	83.5	5476.00	262.32
69.5	3793.67	218.34	84	5541.78	263.89
70	3848.46	219.91	84.5	5607.95	265.46
70.25	3875.99	220.70	85	5674.51	267.04
70.5	3903.63	221.48	85.5	5741.47	268.60
71	3959.20	223.05	86	5808.81	270.17
71.25	3987.13	223.84	86.5	5876.55	271.75
71.5	4015.16	224.62	87	5944.66	273.32
72	4071.51	226.19	87.5	6013.21	274.89
72.25	4099.83	226.98	88	6082.13	276.46
72.5	4128.25	227.75	88.5	6151.44	278.03
73	4185.39	229.34	89	6221.15	279.60
73.25	4214.11	230.12	89.5	6291.25	281.17
73.5	4242.92	230.91	90	6371.64	282.74
74	4300.85	232.48	90.5	6432.62	284.31
74.25	4329.95	233.26	91	6503.89	285.88
74.5	4359.16	234.05	91.5	6573.56	287.46
75	4417.87	235.62	92	6647.62	289.03
75.25	4447.37	236.40	92.5	6720.07	290.60
75.5	4476.97	237.19	93	6792.92	292.17
76	4536.37	238.76	93.5	6866.16	293.74
76.25	4566.36	239.55	94	6939.79	295.31
76.5	4596.35	240.33	94.5	7013.81	296.88
77	4656.63	241.90	95	7088.23	298.45
77.25	4686.92	242.69	95.5	7163.04	300.02
77.5	4717.30	243.47	96	7238.25	301.59
78	4778.37	245.04	96.5	7313.80	303.16
78.25	4809.05	245.83	97	7389.81	304.73
78.5	4839.83	246.61	97.5	7466.22	306.30
79	4901.68	248.19	98	7542.89	307.88
79.25	4932.75	248.97	98.5	7620.09	309.44
79.5	4963.92	249.76	99	7697.70	311.02
80	5026.56	251.33	99.5	7775.63	312.58
80.5	5089.58	252.90	100	7854.00	314.16
81	5153.00	254.47			

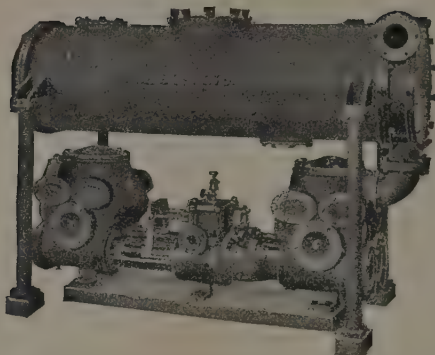


Fig. 99. Worthington Surface Condenser, with Air and Circulating Pump.

TABLE 23.

QUANTITY OF INJECTION WATER FOR JET CONDENSERS.
Injection Temp. 50°. Overflow Temp. 110°.

LOW PRESS. CYLINDER.	WATER PER REV.					
	SINGLE EXP. ENGINES.		DOUBLE EXP. ENGINES.		TRIPLE EXP. ENGINES.	
	Lbs.	Galls.	Lbs.	Galls.	Lbs.	Galls.
20 in. x 36 in.	4.2	.5	3.9	.47	3.6	.43
22 " x 36 "	5.1	.61	4.8	.57	4.4	.53
24 " x 42 "	7.	.84	6.6	.79	6.	.72
26 " x 42 "	8.3	1.	7.8	.93	7.2	.87
28 " x 48 "	11.	1.45	10.4	1.24	9.5	1.14
30 " x 48 "	12.6	1.52	11.7	1.41	10.8	1.3
32 " x 54 "	16.2	1.95	15.	1.81	13.9	1.68
34 " x 54 "	18.3	2.2	17.0	2.05	15.8	1.9
36 " x 60 "	22.8	2.75	21.2	2.55	19.6	2.36
38 " x 60 "	25.5	3.07	23.7	2.85	21.9	2.64
40 " x 66 "	31.	3.73	28.8	3.45	26.7	3.2
44 " x 66 "	37.5	4.51	34.8	4.2	32.2	3.8
48 " x 72 "	48.5	5.84	45.	5.42	41.7	5.
52 " x 72 "	57.	6.89	53.1	6.4	49.2	5.9
56 " x 72 "	66.	7.9	61.5	7.41	57.	6.8
60 " x 72 "	75.6	9.	70.5	8.5	65.3	7.8
64 " x 72 "	85.	10.	80.	9.6	74.	8.9

TABLE 24.

SIZE OF AIR-PUMPS—SINGLE ACTING.
One Stroke of Pump per Rev. of Engine.

LOW PRESS. CYLINDER.	SIZE OF PUMP.		
	SINGLE EXP. ENGINES.	DOUBLE EXP. ENGINES.	TRIPLE EXP. ENGINES.
DIA. STROKE.	DIA. STROKE.	DIA. STROKE.	DIA. STROKE.
20 in. x 36 in.	15½ in. x 8 in.	15 in. x 8 in.	14½ in. x 8 in.
22 " x 36 "	15¼ " x 10 "	16½ " x 8 "	16¼ " x 8 "
24 " x 42 "	17¾ " x 10 "	17¼ " x 10 "	16½ " x 10 "
26 " x 42 "	19½ " x 10 "	18¾ " x 10 "	18 " x 10 "
28 " x 48 "	22½ " x 10 "	21½ " x 10 "	20¾ " x 10 "
30 " x 48 "	21¾ " x 12 "	22¾ " x 10 "	22 " x 10 "
32 " x 54 "	24¾ " x 12 "	24 " x 12 "	23 " x 12 "
34 " x 54 "	26¼ " x 12 "	25½ " x 12 "	24½ " x 12 "
36 " x 60 "	29¼ " x 12 "	28¼ " x 12 "	27¼ " x 12 "
38 " x 60 "	31 " x 12 "	30 " x 12 "	28¾ " x 12 "
40 " x 66 "	34¾ " x 12 "	33 " x 12 "	31¾ " x 12 "
44 " x 66 "	33½ " x 15 "	32½ " x 15 "	34¾ " x 12 "
48 " x 72 "	38 " x 15 "	37 " x 15 "	35½ " x 15 "
52 " x 72 "	41½ " x 15 "	40 " x 15 "	38½ " x 15 "
56 " x 72 "	44½ " x 15 "	43 " x 15 "	42 " x 15 "
60 " x 72 "	47¾ " x 15 "	46¼ " x 15 "	44½ " x 15 "
64 " x 72 "	51 " x 15 "	49½ " x 15 "	47 " x 15 "

MULTIPLE CYLINDER ENGINES.

As has been already explained, a compound engine is one in which the steam is made to do work in two or more cylinders, and the secret of success in this type of engine is due to three factors, viz., (1) a high initial pressure, (2) the expansion of the steam to the greatest extent, and (3) reducing as much as possible the losses caused by cylinder condensation. Prof. Thurston has wisely said that "Maximum expansion, as nearly adiabatic as practicable, is the secret of maximum efficiency."

HORIZONTAL—VERTICAL.

Of the several types of multiple expansion engines, the two cylinder or double expansion engine appears to be best adapted to central power station service, owing to the excessive load variation. Figure 95 shows the horizontal vertical type. It has many advantages, especially in large units. 1st, the low pressure cylinder can be arranged vertically and thereby avoid the excessive friction due to the weight of so large a piston; 2nd,

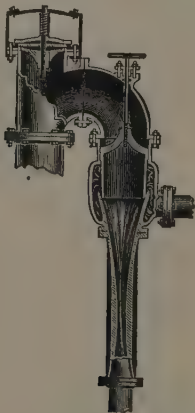


Fig. 100. Siphon Condenser.

the cylinders being arranged one vertical and the other horizontal and both acting upon the same crank pin, gives four impulses for each revolution of the crank. The cut represents a pair of such engines with two cranks, the cranks being keyed to one shaft and at right angles to each other make eight impulses for each revolution, a still greater improvement in turning effect. This type of engine is particularly valuable in electric

generators, with the armature between the engines, and makes a very compact and desirable arrangement.

CROSS COMPOUND.

A cross compound engine consists of two cylinders, one high pressure, and the other low pressure. Each cylinder has its own connecting rod and crank, the cranks being set at opposite ends of the main engine shaft, and at an angle of 90° to each other. The two cylinders are connected by piping, and there is generally a receiver between them, into which the high pressure cylinder exhausts, and is held in reserve until the opening of the low pressure admission valve. Figure 96 shows a cross compound engine, the receiver being underneath the floor. The power unit shown in figure 95 may be considered as consisting of two cross compound engines.

TANDEM COMPOUND.

In the tandem compound, the two cylinders are arranged tandem to each other, as shown in figure 97. The advantage claimed for this type of compound engine is that it gives a much shorter and more direct route for the exhaust steam, in its passage from the high to the low pressure cylinders.

Two Low Pressure Cylinders. Where large units are required, it frequently happens that the low pressure cylinder figures beyond the capacity of the station for handling the work. To meet this limitation, two low pressure cylinders, each of one-half the total area, may be employed; both cylinders being connected with one receiver, three cranks are employed, one in the center and one on each end of same shaft and keyed at 120° to each other.

TABLE 25.

NUMBERS, THEIR SQUARE ROOTS AND CUBE ROOTS.

Square Root.	No.	Cube Root.	Square Root.	No.	Cube Root.	Square Root.	No.	Cube Root.	Square Root.	No.	Cube Root.
3.16	10.	2.15	4.24	18.	2.62	5.10	26.	2.96	5.83	34.	3.24
3.19	10.2	2.16	4.26	18.2	2.63	5.12	26.2	2.96	5.84	34.2	3.24
3.22	10.4	2.18	4.28	18.4	2.64	5.14	26.4	2.97	5.86	34.4	3.25
3.25	10.6	2.19	4.30	18.6	2.64	5.16	26.6	2.98	5.87	34.6	3.26
3.28	10.8	2.20	4.33	18.8	2.65	5.18	26.8	2.99	5.89	34.8	3.26
3.31	11.	2.22	4.35	19.	2.66	5.19	27.	3.00	5.91	35.	3.27
3.34	11.2	2.24	4.38	19.2	2.67	5.21	27.2	3.01	5.92	35.2	3.27
3.37	11.4	2.25	4.40	19.4	2.68	5.23	27.4	3.01	5.94	35.4	3.28
3.40	11.6	2.27	4.43	19.6	2.69	5.25	27.6	3.02	5.96	35.6	3.28
3.43	11.8	2.28	4.45	19.8	2.70	5.27	27.8	3.03	5.98	35.8	3.29
3.46	12.	2.29	4.47	20.	2.71	5.29	28.	3.03	6.00	36.	3.30
3.49	12.2	2.30	4.50	20.2	2.72	5.30	28.2	3.04	6.01	36.2	3.30
3.52	12.4	2.32	4.52	20.4	2.72	5.32	28.4	3.04	6.03	36.4	3.31
3.55	12.6	2.33	4.54	20.6	2.73	5.34	28.6	3.05	6.04	36.6	3.32
3.58	12.8	2.34	4.56	20.8	2.74	5.36	28.8	3.06	6.06	36.8	3.32
3.60	13.	2.35	4.58	21.	2.75	5.38	29.	3.07	6.08	37.	3.33
3.63	13.2	2.37	4.60	21.2	2.76	5.39	29.2	3.07	6.09	37.2	3.33
3.66	13.4	2.38	4.63	21.4	2.77	5.41	29.4	3.08	6.11	37.4	3.34
3.69	13.6	2.39	4.65	21.6	2.78	5.43	29.6	3.08	6.12	37.6	3.34
3.71	13.8	2.40	4.67	21.8	2.79	5.45	29.8	3.09	6.14	37.8	3.35
3.74	14.	2.41	4.69	22.	2.80	5.47	30.	3.10	6.16	38.	3.36
3.76	14.2	2.42	4.71	22.2	2.80	5.49	30.2	3.10	6.17	38.2	3.37
3.79	14.4	2.43	4.73	22.4	2.81	5.50	30.4	3.11	6.19	38.4	3.37
3.82	14.6	2.44	4.75	22.6	2.82	5.52	30.6	3.12	6.20	38.6	3.38
3.85	14.8	2.45	4.77	22.8	2.83	5.54	30.8	3.13	6.22	38.8	3.38
3.87	15.	2.46	4.79	23.	2.84	5.56	31.	3.14	6.24	39.	3.39
3.90	15.2	2.48	4.81	23.2	2.84	5.58	31.2	3.14	6.25	39.2	3.39
3.92	15.4	2.49	4.83	23.4	2.85	5.60	31.4	3.15	6.27	39.4	3.40
3.95	15.6	2.50	4.85	23.6	2.86	5.61	31.6	3.16	6.28	39.6	3.41
3.98	15.8	2.51	4.87	23.8	2.87	5.63	31.8	3.17	6.30	39.8	3.41
4.00	16.	2.52	4.89	24.	2.88	5.65	32.	3.17	6.32	40.	3.42
4.03	16.2	2.53	4.9	24.2	2.88	5.67	32.2	3.18	6.33	40.2	3.42
4.05	16.4	2.54	4.92	24.4	2.89	5.68	32.4	3.18	6.35	40.4	3.43
4.08	16.6	2.55	4.95	24.6	2.90	5.70	32.6	3.19	6.36	40.6	3.43
4.10	16.8	2.56	4.97	24.8	2.91	5.72	32.8	3.19	6.38	40.8	3.44
4.12	17.	2.57	5.00	25.	2.92	5.74	33.	3.20	6.40	41.	3.45
4.14	17.2	2.58	5.02	25.2	2.92	5.76	33.2	3.20	6.41	41.2	3.45
4.17	17.4	2.59	5.04	25.4	2.93	5.77	33.4	3.21	6.43	41.4	3.46
4.19	17.6	2.60	5.06	25.6	2.94	5.79	33.6	3.22	6.45	41.6	3.46
4.22	17.8	2.61	5.08	25.8	2.95	5.81	33.8	3.23	6.46	41.8	3.47

TRIPLE EXPANSION ENGINE.

In the triple expansion engine the steam is expanded successively in three cylinders, each larger in diameter than its predecessor. There is first the high pressure cylinder, second the intermediate cylinder, and third the low pressure cylinder, from which the steam exhausts into the condenser. Very high initial pressure (200 to 225 lbs. per sq. in.) is necessary with this type of engine, as well as the quadruple expansion engine consisting of four cylinders, in order to get good efficiency. The two latter types of multiple expansion engines, viz., the triple and quadruple expansion are best adapted to pumping station work, rather than to the high speeds, and variable loads of the central power station. Owing to present day limitations on boiler pressure the most desirable number of expansions in each cylinder of the different types of condensing engines should be about as given in table 26.

TABLE 26.

NUMBER OF EXPANSIONS.

TYPE.	Initial Steam Pressure. Pounds Abso.	Total Expansions.	EXPANSIONS IN EACH CYLINDER.			
			1st.	2nd.	3rd.	4th.
Single expansions	65	7	7.			
Double expansions	145	22	4.8	4.6		
Triple expansions	185	30	3.2	3.1	3.0	
Quadruple expansions	265	48	2.7	2.65	2.6	2.55

LOAD VARIATION.

One of the worst enemies to economy in the use of steam is a continual variation in load, and the electric railway may be classed among the most unfavorable so far as fluctuations in load is considered.

In a general way the conditions favorable to best economy are:

High Pressure Steam, Superheated.

Proper Ratio of Cylinder Areas.

High Piston Speed.

Low Terminal Pressure.

Good Vacuum, 27 or 28 inches.

Uniform Load (if possible).

Long Stroke.

Steam Jackets.

Uniform Boiler Pressure.

THE STEAM JACKET.

Authorities differ as to the advantages derived from the steam jacket for the low pressure cylinder of a compound engine. There is no doubt that in the case of an engine furnishing power for shop purposes during working hours only, which implies that the service is not continuous, or rather that the engine is in service only ten or twelve hours out of twenty-four, the steam jacket is of great benefit in keeping the cylinder and valve chest warm, and thus preventing the severe strains which would result from the contraction and expansion of the metal. The weight of steam per indicated horse power per hour that is condensed in the jacket varies from 1.7 lb. to 3.8 lb., the average being about 2.3 lb.; while the

economy of the thoroughly steam-jacketed cylinder over the jacketless one varies from 3 lb. to as high as 7.9 lb. of steam per I. H. P. per hour, or an average of about four lbs. less steam consumed per H. P. H. by the use of the jacket, after deducting the weight of steam consumed in the jacket. Judging from all the authorities whom the writer has been able to consult, and also from his own practical experience along this line, it seems plain that an actual saving of from 5 to 15 per cent in the consumption of steam can be effected by the judicious use of the steam jacket. In other words, if an engine with unjacketed cylinders consumes steam at the rate of 20 lbs., per H. P. H. the same engine with its cylinders jacketed would develop the same amount of power with a consumption of only 16 lbs. or 17 lbs. per H. P. H. besides the advantage of having the cylinders always warm and ready for operation.

Figure 101 shows a general view of the engine room of the recently erected central station of the Pacific Light and Power Co. at Redondo, California. The following brief description of the engine plant is condensed from the November, 1908, issue of *Power*:

There are three McIntosh & Seymour compound condensing engines, each unit comprising one vertical and one horizontal cylinder and the same arrangement of low-pressure cylinders. As usual with this make, a superheating receiver is provided between the high-pressure and low-pressure cylinder of each unit, superheated steam being supplied to the heating coils at boiler pressure. The condensation from these coils is almost as hot as the water in the boilers. The engines are 34 and 70 inches bore by 56 inches stroke, and are equipped with the standard McIntosh & Seymour gridiron valves



Fig. 101. View of Engine Room of Pacific Light and Power Company's station.

and valve gear, except that the governor, which is also the standard type used by the builders, controls the point of cutoff through auxiliary valves on both high-pressure and low-pressure cylinders, the main admission valves having fixed travel. The governor is equipped with a delaying dash pot for facilitating parallel operation. The dash pot retards the governor weights for a small fraction of a second, thereby preventing any oscillation of two or more units from becoming amplified.

Each engine is also fitted with an electrically operated speed-changing mechanism applied to the governor; this is operated by a small motor which is controlled from the switchboard gallery. This arrangement allows the engine to be speeded up or down four or five revolutions while running, to bring the generator into synchronism, and is also used to divide the load between engines.

Metallic packing is used on all piston rods and valve rods. The engine cylinders are supplied with oil by force-feed oil pumps.

Previous to acceptance of the plant a demonstration was made which showed that a unit could be successfully operated with one, two or three of the four cylinders; also with one end of any cylinder out of commission.

The main engine bearings and guides are provided with water jackets for cooling, and fresh water is circulated through the jackets by small pumps. The discharge water from the jackets is cooled by means of a small surface condenser; the fresh water for cooling the bearings flows within the tubes and salt water is circulated around the tubes, being taken from condensing-water suction mains.

THE CONDENSING EQUIPMENT.

The plant contains six Wheeler "Admiralty" surface condensers, one being connected directly to each low-pressure cylinder of each main engine. Each condenser

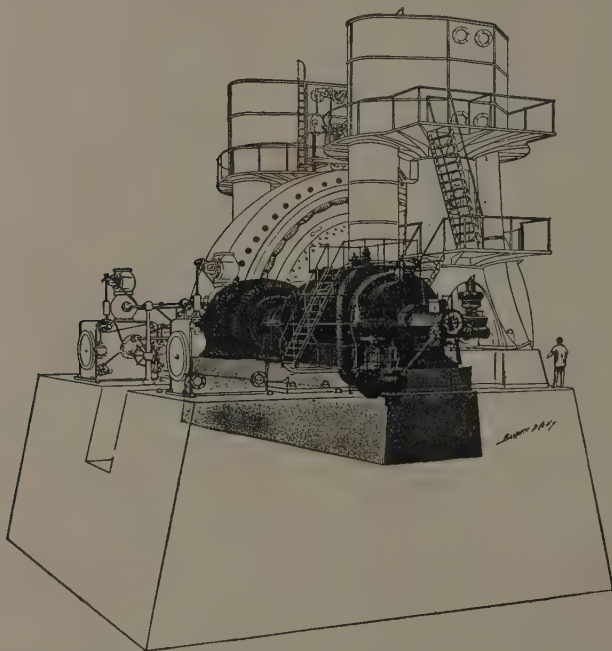


Fig. 102. Comparison of Space Occupied by same Kilowatts of Engine and Turbine Unit.

contains 5,200 square feet of cooling surface; the tubes are of brass, $\frac{3}{4}$ inch outside diameter, with walls No. 18 B. W. G. in thickness. In addition to the regular

cooling surface of each condenser, there is provided in the upper compartment a series of tubes comprising what is termed a Volz heater. After being discharged from the air pump the condensed water passes through this heater under atmospheric pressure and is heated to within a few degrees of the temperature of the exhaust steam surrounding the heater tubes. The make-up water is also brought in at this point, securing the benefit of heat which would otherwise be wasted.

The provision of two condensers on each unit, with cross-connected exhaust passages, is a convenient feature. Should it become necessary for one condenser to be cut out of service, the exhaust steam from both low-pressure cylinders may be led to the remaining condenser, making non-condensing operation unnecessary, except for a failure of circulating water, or the breakdown of both condensers at one time. Actual operation shows but a slight falling off in vacuum due to this doubling of the load on one condenser. Eliminating air leaks in exhaust pipes and minimizing the loss of vacuum due to pipe turns has been given very careful consideration in all its details, the vacuum joints being as carefully made as in high-pressure steam piping. In regular operation with ample water at 70 degrees Fahrenheit a vacuum of from 28, to 28½ inches is easily maintained.

Each condenser is provided with an Edwards triplex suction valveless air pump, motor driven. Each air pump discharges into a vertical stand pipe to eliminate entrained air, after which the water subsequently used as boiler feed passes through the Volz heater in the upper portion of the condenser shell, as previously mentioned. Each of the Edwards pumps has cylinders 16 inches in diameter with 10 inches stroke. The condensed

steam and non-condensable vapors flow continuously by gravity from the condenser to the base of the pump and are removed by the conical bucket or piston peculiar to this pump. The pump is driven by a 25-horsepower General Electric induction motor mounted on an extended base and geared to the pump through a rawhide pinion; only one-fifth of the rated power of the motor is required during normal conditions of load. The motors receive current from the generator of the unit to which they belong; consequently, the air pumps start up gradually and automatically with the main engine.

A siphon system of suction and discharge pipes for circulating water was adopted by reason of the location of plant, the use of a conduit or canal system being impossible. There are two 50-inch suction mains and one discharge line of the same size, supported on a pile wharf carried out to deep water at low tide. The suction pipes are primed by means of a large motor-driven vacuum pump operating through a 40-foot standpipe, which prevents the carrying of water into the vacuum-pump cylinders. When once started, suction is maintained by three engine-driven centrifugal circulating pumps, located in the pump pit, there being one pump for each unit. Each pump has separate connections to both suction lines and all pumps discharge into a common supply main having inlet connections to all condensers; all condenser outlets are connected to one return main. The flexibility of this arrangement is apparent. Any pump may be used to supply any set of condensers; although when two or more pumps are running, their operation is necessarily in parallel. The regulation of relative amounts of water through various condensers is accomplished by the condenser discharge

valves; the engines are speeded up or down, and the number of engines in service is increased or decreased as load conditions require.

Each of the circulating pumps has a 20-inch discharge, and two 15-inch suctions. Each pump is provided with an out board bearing of the ring oiling type, and the pump and bearing are mounted on an extension of the engine base. Direct connected to each pump there is a $9 \times 17\frac{1}{2} \times 12$ Harrisburg tandem compound non-condensing piston valve engine. The limiting speed of the engine is controlled by a throttling governor, but an adjustable eccentric is provided for adjusting the point of cut-off to obtain maximum economy.

The Interborough Railway plant in New York furnishes some interesting details regarding the equipment of a modern power station, in that it is supplied with both reciprocating steam engines, and steam turbines. Figure 103 is presented merely as a comparison of the amount of space occupied by power units of the above mentioned types, having equal capacity.

The Interborough plants in New York are both large. Fig. 103 shows the set of 5000 K. W., 11,000 volts, 263 ampere, 25 cycle, 3 phase alternators run by 7,500 H. P. marine type engines at 75 revolutions per minute. These are in the 59th Street plant. The same type of generator is installed in the 74th Street plant.

ENGINE OPERATION.

In treating upon the operation of steam engines, considerable space will first be devoted to the use of the indicator, as applied to reciprocating engines, owing to the fact that it is practically impossible to properly ad-

just the valves of an engine without the use of the indicator as a guide that will accurately portray the action of the valves when under steam.

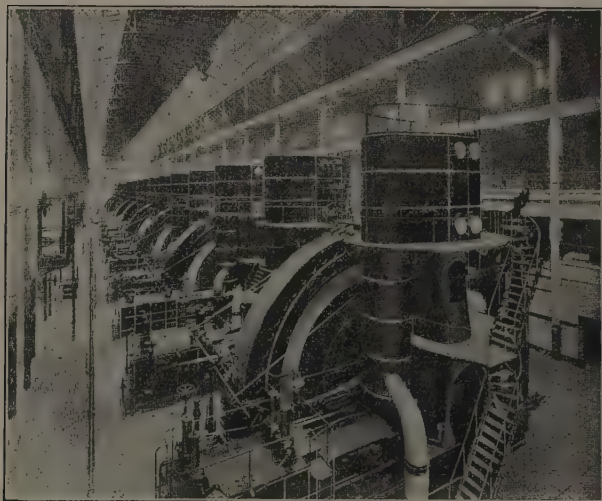


Fig. 103. Set of 5,000 K. W. Engine Alternators, Interborough Plant, New York.

THE INDICATOR.

One of the greatest aids to the economical operation of the steam engine is the indicator, and it is the privilege of every engineer to have at least an elementary, if not a thorough, knowledge of its principles and working. The time devoted to the study of the indicator, and in its application to the engine, is time well spent,

and in the end will well repay the student of steam engineering.

Inventor. The indicator was invented and first applied to the steam engine by James Watt, whose restless genius was not satisfied with a mere outside view of his engine as it was running, but he desired to know more about the action of the steam in the cylinder, its pressure at different portions of the stroke, the laws governing its expansion after being cut off, etc. Watt's indicator, although crude in its design and construction, contained embodied within it all of the principles of the modern instrument.

Principles. These principles are:

First. The pressure of the steam in the engine cylinder throughout an entire revolution, against a small piston in the cylinder of the indicator, which in turn is controlled or resisted in its movement by a spring of known tension, so as to confine the stroke of the indicator piston within a certain small limit.

Second. The stroke of the indicator piston is communicated by a multiplying mechanism of levers and parallel motion to a pencil moving in a straight line. The distance through which the pencil moves being governed by the pressure in the engine cylinder and the tension of the spring.

Third. By the intervention of a reducing mechanism and a strong cord, the motion of the piston of the engine throughout an entire revolution is communicated to a small drum attached to and forming a part of the indicator. The movement of the drum is rotative and in a direction at right angles to the movement of the pencil. The forward stroke of the engine piston causes the drum to rotate through part of a revolution and at

the same time a clock spring connected within the drum is wound up. On the return stroke the motion of the drum is reversed and the tension of the spring returns the drum to its original position and also keeps the cord taut.

To the outside of the drum a piece of blank paper of suitable size is attached and held in place by two clips. Upon this paper the pencil in its motion up and down traces a complete diagram of the pressures and other interesting events transpiring within the engine cylinder during the revolution of the engine. In fact the diagram traced upon the paper is the compound result of two concurrent movements. First, that of the pencil caused by the pressure of the steam against the indicator piston; second, that of the paper drum caused by, and coincident with, the motion of the engine piston. The upper end of the indicator cylinder is always open to the atmosphere, the steam acting only upon the underside of the small piston, and when the cock connecting the cylinders of the engine and indicator is closed, both ends of the indicator cylinder are open to atmospheric pressure, and the pencil then stands at its neutral position. If now the pencil is held against the paper and the drum rotated either by hand or by connecting it with the cord, a horizontal line will be traced. This line is called the atmospheric line, meaning the line of atmospheric pressure, and it is a very important factor in the study of the diagram.

If the engine is a non-condensing engine the pencil in tracing the diagram will, or at least should, not fall below the atmospheric line at any point, but will on the return stroke trace a line called the line of back pressure at a distance more or less above the atmospheric

line and very nearly parallel with it. If the engine is a condensing engine the pencil will drop below the atmospheric line while tracing the line of back pressure on the diagram, and the distance this line is below the atmospheric line will depend upon the number of inches of vacuum in the condenser.

As before stated, the length of stroke of the indicator piston, and the pencil movement as well is controlled by a spiral steel spring which acts in resistance to the pressure of the steam. These springs are made of different tensions in order to be suitable to different steam pressures and speeds, and are numbered 20, 40, 60, etc., the number meaning that a pressure per square inch in the engine cylinder corresponding to the number on the spring will cause a vertical movement of the pencil through a distance of one inch. Thus, if a number 20 spring is used and the pressure in the cylinder at the commencement of the stroke is 20 lbs. per square inch, the pencil will be raised one inch, or if the pressure is 30 lbs., the pencil will travel $1\frac{1}{2}$ in., and if there is a vacuum of 20 in. in the condenser, the pencil will drop $\frac{1}{2}$ in. below the atmospheric line for the reason that 20 in. of vacuum corresponds to a pressure of about 10 lbs. less than atmospheric pressure or an absolute pressure of about 4 lbs. If a 60 spring is used a pressure of 60 lbs. in the engine cylinder will be required to raise the pencil one inch, or 90 lbs. to raise it $1\frac{1}{2}$ in.

The Ashcroft Manufacturing Co. of New York, makers of the well known Tabor indicator, have recently introduced a new feature in indicator work by connecting the spring on top of the cylinder and in plain view of the operator. This arrangement removes the spring from the influence of direct contact with the steam, and

it is subject only to the temperature of the surrounding atmosphere. It is claimed that as a result of this the accuracy of the spring is insured, and that no allowance

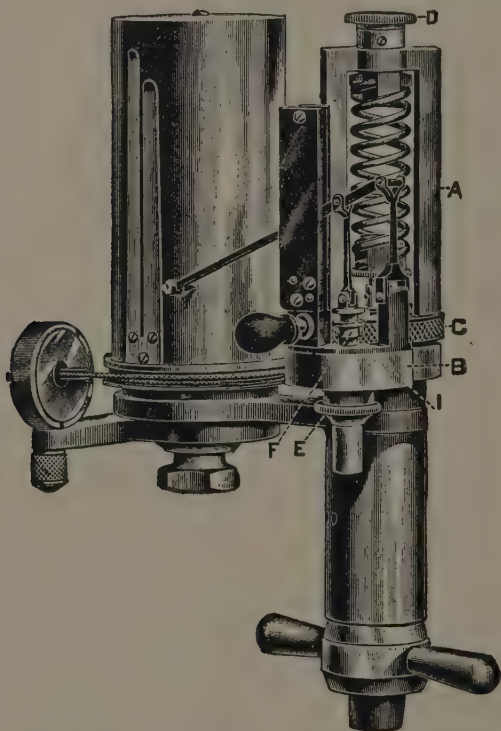


Fig. 104. Improved Tabor Indicator with Outside Connected Spring
Ashcroft Mfg. Co., N. Y.

need to be made in its manufacture for expansion caused by the high temperature to which it is subject when

located within the cylinder. Another good feature of this design is, that the spring can be easily removed without disconnecting any one part of the instrument in case it is desired to change springs. A cut of the improved instrument is herewith presented. See figure 104.

Fig. 104a is a sectional view of the American Thompson improved indicator. Fig. 105 shows the spring. Fig. 106 is a three way cock for attaching the indicator to the cylinder.

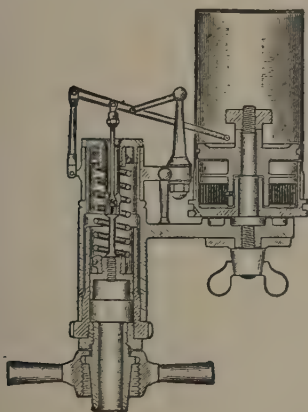


Fig. 104a.



Fig. 105.

Reducing Mechanism. Probably the only practically universal mechanism for reducing the motion of the crosshead is the reducing wheel, a device in which, by the employment of gears and pulleys of different diameters, the motion is reduced to within the compass of the drum, and the device is applicable to almost any make of

engine, whether of high or low speed. Some makers of indicators attach the reducing wheel directly to the indicator, thus producing a neat and very convenient arrangement. Fig. 108 shows the indicator complete with reducing wheel attached.

One of the most accurate and easily applied devices for reducing the motion of the piston is the wooden pendulum in its various forms. (See Figs. 109, 110 and 111.) It consists of a flat strip of pine or other light wood of a length not less than one and a half times the



Fig. 106.

CROSBY
INDICATOR
SPRING

Fig. 107.

stroke of the engine, and if made longer it will be better. It should be from $\frac{3}{4}$ to $\frac{7}{8}$ in. thick and have an average width of about 4 in. If the engine to be indicated is horizontal the bar or pendulum is to be pivoted at a fixed point directly above and in line with the side of the crosshead, as that is generally the most convenient

point of attachment. The pivot can be fixed to a permanent standard bolted to the frame of the engine (Fig. 112), or it may be secured to the ceiling of the room or even to a post fastened to the floor. If the engine is vertical the bar can be pivoted to the wall of the room or a strong post firmly secured to the floor. The connection with the crosshead is best accomplished by means of a

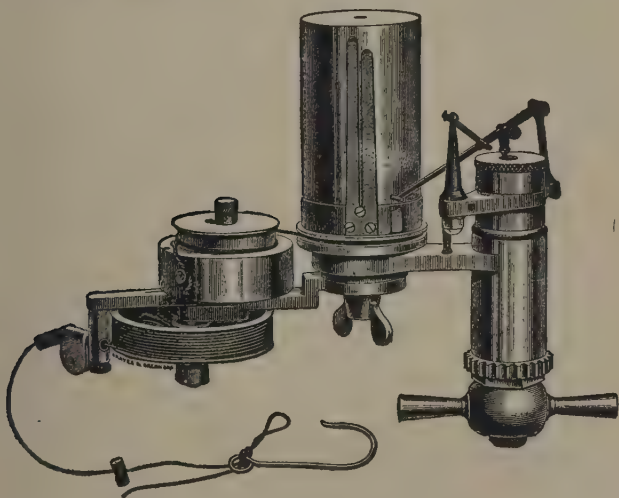


Fig. 108.

short bar or link. A convenient length for this bar is one-half the stroke of the engine. To locate the correct point for the pivot, assuming the length of the short bar to be one-half the length of the stroke, proceed as follows:

Place the engine on the center with the crosshead at the end of the stroke towards the crank. Then having

previously bored a hole for the pivot in one end of the pendulum bar and in the other end a hole for connecting with the link, suspend the pendulum by a temporary pin, as a large wood screw, directly above and in line with the stud or bolt hole which has previously been tapped into the crosshead at any convenient point. The

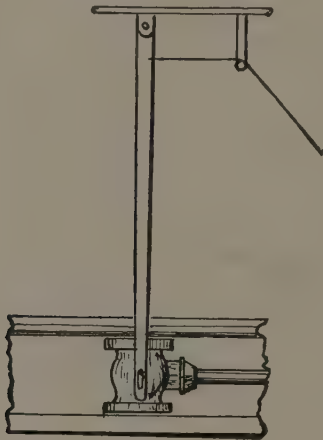


Fig. 109.

pendulum should be temporarily suspended at such a height that when it hangs perpendicular the hole in its lower end will line up accurately with the hole or stud in the crosshead. Now swing the pendulum in either direction a distance equal to the length of the link (one-half the stroke of the engine) from the crosshead connection and note the distance that the bottom hole is above a straight edge laid horizontal and in line with the center of the stud in the crosshead. This will give the

total vibration of the free end of the link from a line parallel with the line of the engine and the permanent location of the pivot should be one-half of this distance below the temporary point of suspension. This will allow the link to vibrate equally above and below the center of its connection with the crosshead. Fig. 113 shows a complete connection of this character.

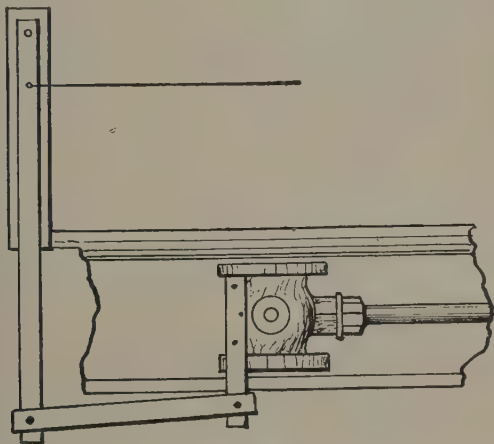


Fig. 110.

Sometimes the end is slotted and thus directly connected to the stud in the crosshead, dispensing with the link. In this case it is necessary to locate the pivot at a point perpendicular to the center of travel of the stud in the crosshead. (See Fig. 109.) The link connection is to be preferred, however. The cord can be attached to the pendulum at a point near the pivot which will give the desired length of diagram. This point can be determined by multiplying the length of the pendulum by the

desired length of diagram and dividing the product by the stroke. For convenience these terms should be expressed in inches. Thus, assume stroke of engine to be 48 in., length of pendulum $1\frac{1}{2}$ times length of stroke = 72 in. Desired length of diagram 3 in. Then $72 \times 3 \div 48 = 4.5$ in., which is the distance from center of

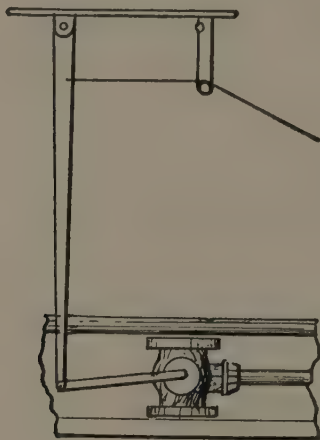


Fig. 111.

pivot to point of connection for the cord. This can be either a small hole bored through the pendulum or a wood screw to which the cord can be attached. From this point the cord should be led over a guide pulley located at such height that when the pendulum is vertical the cord will leave it at right angles. After leaving the guide pulley the cord can be carried at any angle desired.

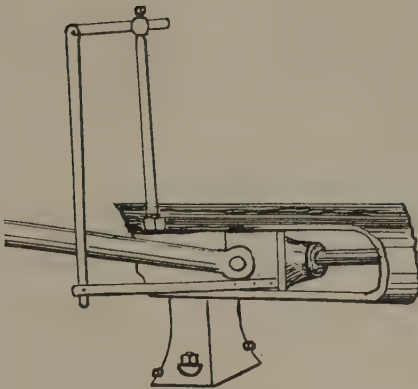


Fig. 112.

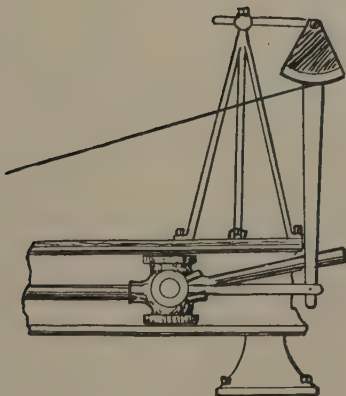


Fig. 113.

The Brumbo Pulley. Another method of connecting the cord to the pendulum is to run the cord over a grooved segment, called a Brumbo pulley, connected with the pivoted end of the pendulum (Fig. 114), but with this arrangement, owing to the curved travel of the pendulum, there is greater liability to distortion of the

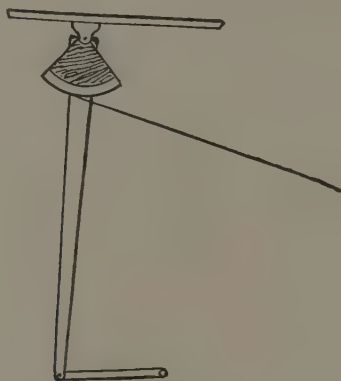


Fig. 114.

diagram than in the first method. In case it is desired to use the Brumbo pulley, the radius of the segment can be found by the same process as that used for finding the point for connecting the cord directly to the pendulum.

One of the neatest and most easily applied devices for reducing the motion of the crosshead is the pantograph. (See Fig. 115.) No dimensions are essential except that it shall be made reasonably strong of some light, tough variety of wood, and that the pins and holes be nicely fitted to each other so that while the movement may be free, there shall at the same time not be too much lost mo-

tion. The pantograph should be of such capacity that it will just close up nicely when the engine is at mid stroke and open out nicely when at its extreme travel. The two ends, C and D, are each to be fitted with a pin extending through far enough so that pin C can be hooked into a hole or socket on the crosshead, while pin D rests in a socket in the top of a post secured to the floor at a point opposite the center of travel of the crosshead and of

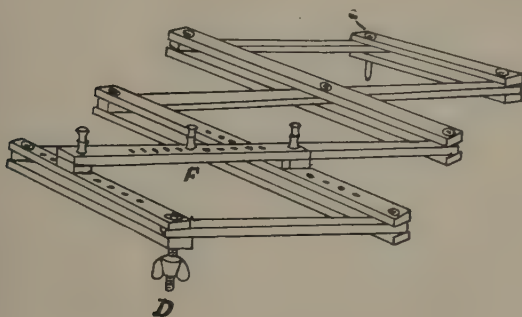


Fig. 115.

such height as will allow the pantograph to lie in a horizontal position. Also the distance of the post from the guides must be adjusted so as to allow the device to close up at mid stroke and open out at full stroke without any straining of the parts. The point F of connection for the cord will always have a motion parallel with, and simultaneous with, that of the crosshead; the pin to which the cord is attached can be set in any one of the holes that will give the desired length for the diagram. The motion given by this device is accurate, although it may become necessary in some cases, especially with long

stroke engines, to introduce a guide pulley to carry the cord from the pantograph.

Attaching the Indicator. The cylinders of most engines at the present time are drilled and tapped for indicator connections before they leave the shop, which is

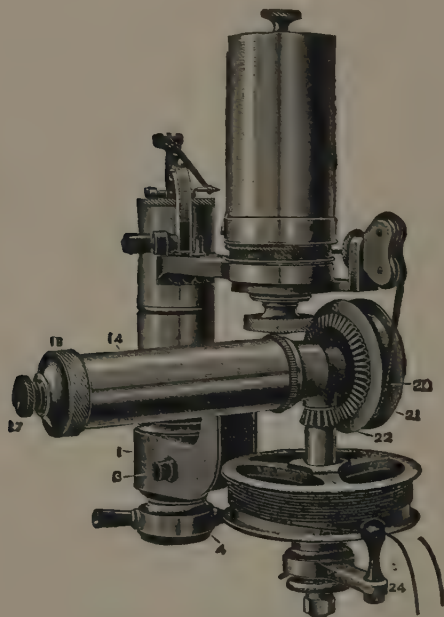


Fig. 116. Crosby Reducing Wheel.

eminently proper, as no engine builder, or purchaser either, should be satisfied with the performance of a new engine until after it has been accurately tested and adjusted with the indicator.

The main requirements in these connections are that the holes must not be drilled near the bottom of the cylinder where water is likely to find its way into the pipes, neither should they be in a location where the inrush of steam from the ports will strike them directly, nor where the edge of the piston is liable to partly cover them when at its extreme travel. An engineer before he undertakes to indicate an engine should satisfy himself that all these requirements are fulfilled. Otherwise he is not likely to obtain a true diagram. The cock supplied with the indicator is threaded for one-half inch pipe and unless the engine has a very long stroke it is the practice to bring the two end connections together at the side or top of the cylinder and at or near the middle of its length, where they can be connected to a three way cock. The pipe connections should be as short and as free from elbows as possible in order that the steam may strike the indicator piston as nearly as possible at the same moment that it acts upon the engine piston.

The work of taking diagrams is very much simplified by having both ends of the cylinder connected to one common tee, or a three way cock as above described, but for long stroke engines there should be two indicators, one for each end and the diagrams should be taken simultaneously if it is desired to adjust the valves by the indicator. In this case an assistant would be required to manipulate one of the instruments.

The pipes should always be thoroughly blown out, by allowing the steam to blow through the open cock during several revolutions of the engine, before connecting the indicator. If this is not done there is a moral certainty that grit and dirt will get into the cylinder of the indi-

cator, where the presence of the least atom of grit will cause the delicate instrument to work badly.

Selecting a Spring. The proper number of spring to use depends upon the boiler pressure in the case of an automatic cut off engine, but for an engine with a fixed cut off and throttling governor the number of the spring to be selected will depend upon the initial pressure in the cylinder. A convenient rule is to select a spring numbered one-half as high as the pressure; for instance, if the boiler pressure is 80 lbs., use a No. 40 spring, which will give a diagram 2 in. in height.

Care of the Instrument. The indicator should be cleaned and oiled both before and after using. The best material for wiping it is a clean piece of old soft muslin of fine texture, as there is not so much liability of lint sticking to, or getting into the small joints. Use good clock oil for the joints and springs, and before taking diagrams it is a good practice to rub a small portion of cylinder oil on the piston and the inside of the cylinder, but when about to put the instrument away these should be oiled with clock oil also. None but the best cord should be used for connecting the paper drum with the reducing motion, as a cord that is liable to stretch will cause trouble. Suitable cord and also blank diagrams can generally be secured from firms manufacturing, and selling indicators. After the indicator has been screwed on to the cock connecting with the pipe, the cord must be adjusted to the proper length before hooking it on to the drum. This must be done while the engine is running, by taking hold of the loop on the cord connected with the reducing motion with one hand, and with the other hand grasp the hook on the short cord attached to the drum, then by holding the two ends near each other

during a revolution or two it will be seen whether the long cord needs to be shortened or lengthened.

The length of the diagram is determined by the point of connection of the cord to the pendulum as has been heretofore explained. Care should be exercised in placing the paper on the drum to see that it is stretched tight and firmly held by the clips. The pencil point having been first sharpened by rubbing it on a piece of fine emery cloth or sand paper should be adjusted by means of the pencil stop with which all indicators should be provided, so that it will have just sufficient bearing against the paper to make a fine, plain mark. If the pencil bears too hard on the paper it will cause unnecessary friction and the diagram will be distorted. The best method of ascertaining this fact and also whether the travel of the drum is equally divided between the stops, is to place a blank diagram on the drum, connect the cord and while the engine makes a revolution hold the pencil against the paper. Then unhook the cord, remove the paper and if the travel of the drum is not divided correctly it can be changed.

Having thus arranged all the preliminary details, place a fresh blank on the drum, being careful to keep the pencil out of contact with it, connect the cord, open the cock admitting steam to the indicator and after the pencil has made a few strokes to allow the cylinder to become warmed up, then gently swing it around to the paper drum and hold it there while the engine makes a complete revolution. Then move the pencil clear of the paper, close the cock and unhook the cord. Now trace the atmospheric line by holding the pencil against the paper while the drum is revolved by hand. This method of tracing the atmospheric line is preferable to that of

tracing it immediately after closing the cock and while the drum is still being moved by the engine, for the reason that there is not so much liability of getting the atmospheric line too high owing to the presence of a slight pressure of steam remaining under the indicator piston for a second or two just after closing the cock; also the line drawn by hand will be longer than one drawn while the drum is moved by the motion of the engine and will therefore be more readily distinguished from the line of back pressure.

Having secured a truthful diagram, it now remains to take as many as are desired, and if the object is to set the valves of the engine, the diagrams from each end of the cylinder should follow each other as quickly as possible in order that the conditions of load and steam pressure may be the same. When the indicator is connected so that diagrams can be taken from both ends without changing it, the above conditions can generally be realized. But if diagrams can only be taken from one end at a time, the only way to arrive at correct conclusions in relation to the adjustment of the valves will be to see that the boiler pressure is practically the same at the time of taking diagrams from either end and that the position of the governor is also the same, assuming that the load on the engine is practically constant. This applies of course to an automatic cut off.

As soon as the diagrams are taken the following data should be noted upon them: The end of the cylinder, whether head or crank; boiler pressure; and time when taken. Other data can be added afterwards. If the engine is an automatic cut off of the Corliss type and the point of cut off on one end does not coincide with the other, the difference can generally be adjusted while the engine is running by changing the length of the rods

extending from the governor to the tripping device. These rods are, or should be, fitted with right and left threads on the ends for this purpose. Any changes in the valves, such as giving them more lead, compression, etc., and which necessitates changing the length of the reach rods connecting them with the wrist plate, will have to be made while the engine is stopped, although with slow speed engines and the exercise of caution it is possible to make alterations in these rods while the engine is running.

DIAGRAM ANALYSIS. In order to facilitate the study and analysis of indicator diagrams, the following definitions of technical terms, some of which have already been explained in another part of this book, are here given.

Absolute pressure. Pressure reckoned from a perfect vacuum. It equals the boiler pressure plus the atmospheric pressure.

Boiler pressure or gauge pressure. Pressure above the atmospheric pressure as shown by the steam gauge.

Initial pressure. Pressure in the cylinder at the beginning of the stroke.

Terminal pressure (T. P.). The pressure that would exist in the cylinder at the end of the stroke provided the exhaust valve did not open until the stroke was entirely completed. It may be graphically illustrated on the diagram by extending the expansion curve by hand to the end of the stroke. It is found theoretically by dividing the pressure at point of cut off by the ratio of expansion. Thus, absolute pressure at cut off = 100 lbs., ratio of expansion = 5; then $100 \div 5 = 20$ lbs., absolute terminal pressure.

Mean effective pressure (M. E. P.). The average

pressure acting upon the piston throughout the stroke minus the back pressure.

Back pressure. Pressure which tends to retard the forward stroke of the piston. Indicated on the diagram from a non-condensing engine by the height of the back pressure line above the atmospheric line. In a condensing engine the degree of back pressure is shown by the height of the back pressure line above an imaginary line representing the pressure in the condenser corresponding to the degree of vacuum in inches, as shown by the vacuum gauge.

Total or absolute back pressure, in either a condensing or non-condensing engine, is that indicated on the diagram by the height of the line of back pressure above the line of perfect vacuum.

Ratio of expansion. The proportion that the volume of steam in the cylinder at point of release bears to the volume at cut off. Thus, if the point of cut off is at one-fifth of the stroke, and release does not take place until the end of the stroke, the ratio of expansion, or in other words, the number of expansions, is 5. When the T. P. is known the ratio of expansion may be found by dividing the initial pressure by the T. P.

Wire drawing. When through insufficiency of valve opening, contracted ports, or throttling governor, the steam is prevented from following up the piston at full initial pressure until the point of cut off is reached, it is said to be wire drawn. It is indicated on the diagram by a gradual inclination downwards of the steam line from the admission line to the point of cut off. Too small a steam pipe from boiler to engine will also cause wire drawing and fall of pressure.

Condenser pressure may be defined as the pressure ex-

isting in the condenser of an engine, caused by the lack of a perfect vacuum. As, for instance, with a vacuum of 25 in. there will still remain the pressure due to the 5 in. which is lacking. This will be about 2.5 lbs.

Vacuum. That condition existing within a closed vessel from which all matter, including air, has been expelled. It is measured by inches in a column of mercury contained within a glass tube a little over 30 in. in height, having its lower end open and immersed in a small open vessel filled with mercury. The upper end of the glass tube is connected with the vessel in which the vacuum is to be produced. When no vacuum exists the mercury will leave the tube and fill the lower vessel. When a vacuum is maintained in the condenser, or other vessel, the mercury will rise in the glass tube to a height corresponding to the degree of vacuum. If the mercury rises to the height of 30 in. it indicates a perfect vacuum, which means the absence of all pressure within the vessel, but this condition is never realized in practice; the nearest approach to it being about 28 in.

For purposes of convenience the mercurial vacuum gauge is not generally used, it having been replaced by the Bourdon spring gauge, although the mercury gauge is used for testing.

The vacuum in a condenser is generally maintained by an air pump, although it can be produced and maintained by the mere condensation of the steam as it enters the condenser by allowing a spray of cold water to strike it. The steam when it first enters the condenser drives out the air and the vessel is filled with steam which, when condensed, occupies about 1,600 times less space than it did before being condensed, hence a partial vacuum is produced.

While the vacuum in a condenser cannot be considered as power at all, yet it occupies the anomalous position of increasing, by its presence, the capacity of the engine for doing work. This is owing to the fact that the atmospheric pressure or resistance which is always ahead of the piston in a non-condensing engine is, in the case of a condensing engine, removed to a degree corresponding to the height of the vacuum, thus making available just so much more of the pressure behind the piston. Thus, if the average steam pressure throughout the stroke is 30 lbs. and there is a vacuum of 26 in. maintained in the condenser, there will be 13 lbs. of resistance per square inch removed from in front of the piston, thus making available $30 + 13 = 43$ lbs. pressure per square inch.

Absolute zero has been fixed by calculation at 461.2° below the zero of the Fahrenheit scale.

Piston displacement. The space or volume swept through by the piston in a single stroke. Found by multiplying the area of piston by length of stroke.

Piston clearance. The distance between the piston and cylinder head when the piston is at the end of the stroke.

Steam clearance, ordinarily termed clearance. The space between the piston at the end of the stroke and the valve face. It is reckoned in per cent of the total piston displacement.

Horse power (H. P.). 33,000 pounds raised one foot high in one minute of time.

Indicated horse power (I. H. P.). The horse power as shown by the indicator diagram. It is found as follows:

Area of piston in square inches $\times$ M. E. P. $\times$ piston speed in feet $\div$ 33,000.

Piston speed. The distance in feet traveled by the piston in one minute. It is the product of twice the length of stroke expressed in feet multiplied by the number of revolutions per minute.

R. P. M. Revolutions per minute.

Net horse power. I. H. P. minus the friction of the engine.

Compression. The action of the piston as it nears the end of the stroke, in reducing the volume and raising the pressure of the steam retained in the cylinder ahead of the piston by the closing of the exhaust valve.

Boyle's or Mariotte's law of expanding gases. "The pressure of a gas at a constant temperature varies inversely as the space it occupies." Thus, if a given volume of gas is confined at a pressure of 50 lbs. per square inch and it is allowed to expand to twice its volume, the pressure will fall to 25 lbs. per square inch.

Adiabatic curve. A curve representing the expansion of a gas which loses no heat while expanding. Sometimes called the curve of no transmission.

Isothermal curve. A curve representing the expansion of a gas having a constant temperature, but partially influenced by moisture, causing a variation in pressure according to the degree of moisture or saturation. It is also called the theoretical expansion curve.

Expansion curve. The curve traced upon the diagram by the indicator pencil showing the actual expansion of the steam in the cylinder.

First law of thermodynamics. Heat and mechanical energy are mutually convertible.

Power. The rate of doing work, or the number of foot pounds exerted in a given time.

Unit of work. The foot pound, or the raising of one pound weight one foot high.

First law of motion. All bodies continue either in a state of rest or of uniform motion in a straight line, except in so far as they may be compelled by impressed forces to change that state.

Work. Mechanical force or pressure cannot be considered as work unless it is exerted upon a body and causes that body to move through space. The product of the pressure multiplied by the distance passed through and the time thus occupied is work.

Momentum. Force possessed by bodies in motion, or the product of mass and density.

Dynamics. The science of moving powers or of matter in motion, or of the motion of bodies that mutually act upon each other.

Force. That which alters the motion of a body, or puts in motion a body that was at rest.

Maximum theoretical duty of steam is the product of the mechanical equivalent of heat, viz., 778 ft. lbs. multiplied by the total heat units in a pound of steam. Thus, in one pound of steam at 212° reckoned from 32° the total heat equals 1,146.6 heat units. Then $778 \times 1,146.6$ equals 892,054.8 ft. lbs. = maximum duty.

Steam efficiency may be expressed as follows:

Heat converted into useful work	and maximum efficiency
Heat expended	

can only be attained by using steam at as high an initial pressure as is consistent with safety and at as large a ratio of expansion as possible. The percentage of efficiency of steam used at atmospheric pressure in a non-expansive engine is very low; as, for instance, the

heat expended in the evaporation of one pound of water at 32° into steam at atmospheric pressure = 1,116.6 heat units, and the volume of steam so generated = 26.36 cu. ft.

One cubic foot of steam at 212° contains energy equal to $144 \times 14.7 = 2,116.8$ ft. lbs., and 26.36 cu. ft. = $2,116.8 \times 26.36 = 55,798.84$ ft. lbs., which divided by the mechanical equivalent of heat, viz., 778 ft. lbs. = 71.72 heat units, available for useful work. The per cent of efficiency therefore is $\frac{71.72 \times 100}{1,116.6} = 6.28$ per cent. But

suppose the initial pressure to have been 200 lbs. absolute, and that the steam is allowed to expand to thirty times its original volume. The heat expended in evaporating a pound of water at 32° into steam at 200 lbs. absolute pressure = 1,198.3 heat units, and the volume of steam so generated = 2.27 cu. ft. The average pressure during expansion would be 29.34 lbs. per square inch and the volume when expanded thirty times would equal $2.27 \times 30 = 68.1$ cu. ft.

One cubic foot of steam at 29.34 lbs. pressure equals $144 \times 29.34 = 4,224.96$ ft. lbs., and 68.1 cu. ft. will equal $4,224.96 \times 68.1 = 287,719.7$ ft. lbs. of energy, which divided by the equivalent, 778, equals 370.2 heat units, and the per cent of efficiency will be $\frac{370.2 \times 100}{1,198.3} = 30.8$ per cent.

Engine efficiency. If the engine is considered merely as a machine for converting into useful work the heat energy in the steam regardless of the cost of fuel, its efficiency may be expressed as follows:

$$\frac{\text{Heat converted into useful work}}{\text{Total heat received in the steam}}$$

Example. Assume an engine to be receiving steam at 95 lbs. absolute pressure, that the consumption of dry steam per horse power per hour equals 20 lbs., that the friction of the engine amounts to 15 per cent, and that the temperature of the feed water is raised from 60° to 170° by utilizing a portion of the exhaust.

In a pound of steam at 95 lbs. absolute there are 1,180.7 heat units, and in a pound of water at 170° there are 138.6 units of heat, but 28.01 of these heat units were in the water at its initial temperature of 60°. Therefore the total heat added to the water by the exhaust steam equals $138.6 - 28.01 = 110.59$ heat units, and the total heat in each pound of steam to be charged up to the engine is $1,180.7 - 110.59 = 1,070.11$, and the total for each horse power developed per hour will be $1,070.11 \times 20 = 21,402.2$ heat units.

A horse power equals 33,000 ft. lbs. per minute, or sixty times 33,000 = 1,980,000 ft. lbs. per hour. From this must be deducted 15 per cent for friction of the engine leaving 1,683,000 ft. lbs. for useful work. Dividing this by the equivalent, viz., 778 ft. lbs., gives 2,163.2 heat units as the heat converted into one horse power of work in one hour, and the percentage of efficiency of the engine will be $\frac{2,163.2 \times 100}{21,402.2} = 10.1$ per cent.

Efficiency of the plant as a whole includes boiler and engine efficiency and is to be figured upon the basis of

$$\frac{\text{Heat converted into useful work}}{\text{Calorific or heat value of fuel}}$$

Horse power constant of an engine is found by multiplying the area of the piston in square inches by the speed of the piston in feet per minute and dividing the product by 33,000. It is the power the engine would

develop with one pound mean effective pressure. To find the horse power of the engine, multiply the M. E. P. of the diagram by this constant.

Logarithms. A series of numbers having a certain relation to the series of natural numbers, by means of which many arithmetical operations are made comparatively easy. The nature of the relation will be understood by considering two simple series, such as the following, one proceeding from unity in geometrical progression and the other from 0 in arithmetical progression:

Geom. series, 1, 2, 4, 8, 16, 32, 64, 128, 256, 512, etc.

Arith. series, 0, 1, 2, 3, 4, 5, 6, 7, 8, 9, etc.

Here the ratio of the geometrical series is 2 and any term in the arithmetical series expresses how often 2 has been multiplied into 1 to produce the corresponding term of the geometrical series. Thus, in proceeding from 1 to 32 there have been 5 steps or multiplications by the ratio 2; in other words, the ratio of 32 to 1 is compounded 5 times of the ratio of 2 to 1. The above is the basic principle upon which common logarithms are computed.

Hyperbolic logarithms. Used in figuring the M. E. P. of a diagram from the ratio of expansion and the initial pressure. Thus, hyperbolic logarithm of ratio of expansion $\div$ 1 multiplied by absolute initial pressure and divided by ratio of expansion = mean forward pressure. From this deduct total back pressure and the remainder will be mean effective pressure. The hyperbolic logarithm is found by multiplying the common logarithm by the constant 2.302585. Table 27 gives the hyperbolic logarithms of numbers usually required in calculations of the above nature.

Table 27.

HYPERBOLIC LOGARITHMS.

No.	Log.	No.	Log.	No.	Log.	No.	Log.
1.01	0.0099	3.10	1.1341	5.20	1.6486	7.30	1.9879
1.05	0.0477	3.15	1.1474	5.25	1.6582	7.35	1.9947
1.10	0.0953	3.20	1.1631	5.30	1.6677	7.40	2.0015
1.15	0.1397	3.25	1.1786	5.35	1.6771	7.45	2.0018
1.20	0.1822	3.30	1.1939	5.40	1.6864	7.50	2.0149
1.25	0.2231	3.35	1.2090	5.45	1.6956	7.55	2.0215
1.30	0.2623	3.40	1.2238	5.50	1.7047	7.60	2.0281
1.35	0.3001	3.45	1.2384	5.55	1.7138	7.65	2.0347
1.40	0.3364	3.50	1.2527	5.60	1.7228	7.70	2.0412
1.45	0.3715	3.55	1.2669	5.65	1.7316	7.75	2.0477
1.50	0.4054	3.60	1.2809	5.70	1.7405	7.80	2.0541
1.55	0.4382	3.65	1.2947	5.75	1.7491	7.85	2.0605
1.60	0.4700	3.70	1.3083	5.80	1.7578	7.90	2.0668
1.65	0.5007	3.75	1.3217	5.85	1.7664	7.95	2.0731
1.70	0.5306	3.80	1.3350	5.90	1.7750	8.00	2.0794
1.75	0.5596	3.85	1.3480	5.95	1.7834	8.05	2.0857
1.80	0.5877	3.90	1.3610	6.00	1.7918	8.10	2.0918
1.85	0.6151	3.95	1.3737	6.05	1.8000	8.15	2.0988
1.90	0.6418	4.00	1.3863	6.10	1.8083	8.20	2.1041
1.95	0.6678	4.05	1.3987	6.15	1.8164	8.25	2.1102
2.00	0.6931	4.10	1.4010	6.20	1.8245	8.30	2.1162
2.05	0.7178	4.15	1.4231	6.25	1.8326	8.35	2.1222
2.10	0.7419	4.20	1.4351	6.30	1.8405	8.40	2.1282
2.15	0.7654	4.25	1.4469	6.35	1.8484	8.45	2.1342
2.20	0.7885	4.30	1.4586	6.40	1.8563	8.50	2.1400
2.25	0.8110	4.35	1.4701	6.45	1.8640	8.55	2.1459
2.30	0.8329	4.40	1.4816	6.50	1.8718	8.60	2.1518
2.35	0.8544	4.45	1.4929	6.55	1.8795	8.65	2.1576
2.40	0.8755	4.50	1.5040	6.60	1.8870	8.70	2.1633
2.45	0.8961	4.55	1.5151	6.65	1.8946	8.75	2.1690
2.50	0.9163	4.60	1.5260	6.70	1.9021	8.80	2.1747
2.55	0.9361	4.65	1.5369	6.75	1.9095	8.85	2.1804
2.60	0.9555	4.70	1.5475	6.80	1.9169	8.90	2.1860
2.65	0.9746	4.75	1.5581	6.85	1.9242	8.95	2.1916
2.70	0.9932	4.80	1.5686	6.90	1.9315	9.00	2.1972
2.75	1.0116	4.85	1.5790	6.95	1.9387	9.05	2.2028
2.80	1.0296	4.90	1.5892	7.00	1.9459	9.10	2.2083
2.85	1.0473	4.95	1.5994	7.05	1.9530	9.15	2.2137
2.90	1.0647	5.00	1.6094	7.10	1.9600	9.20	2.2192
2.95	1.0818	5.05	1.6194	7.15	1.9671	9.25	2.2246
3.00	1.0986	5.10	1.6292	7.20	1.9740	9.30	2.2310
3.05	1.1151	5.15	1.6390	7.25	1.9810	9.35	2.2354

TABLE 27—Continued.

No.	Log.	No.	Log.	No.	Log.	No.	Log.
9.40	2.2407	9.80	2.2824	11.00	2.3379	19.00	2.9444
9.45	2.2460	9.85	2.2875	12.00	2.4449	20.00	2.9957
9.50	2.2513	9.90	2.2925	13.00	2.5536	21.00	3.0445
9.55	2.2565	9.95	2.2976	14.00	2.6596	22.00	3.0910
9.60	2.2618	10.00	2.3026	15.00	2.7163	23.00	3.1355
9.65	2.2670	10.25	2.3273	16.00	2.7751	24.00	3.1790
9.70	2.2721	10.50	2.3514	17.00	2.8332	25.00	3.2199
9.75	2.2773	10.75	2.3749	18.00	2.8903	30.00	3.3782

Steam consumption per horse power per hour. The weight in pounds of steam exhausted into the atmosphere or into the condenser in one hour divided by the horse power developed. It is determined from the diagram by selecting a point in the expansion curve just previous to the opening of the exhaust valve and measuring the absolute pressure at that point. Then the piston displacement up to the point selected, plus the clearance space, expressed in cubic feet, will give the volume of steam in the cylinder, which multiplied by the weight per cubic foot of steam at the pressure as measured will give the weight of steam consumed during one stroke. From this should be deducted the steam saved by compression as shown by the diagram, in order to get a true measure of the economy of the engine. Having thus determined the weight of steam consumed for one stroke, multiply it by twice the number of strokes per minute and by 60, which will give the total weight consumed per hour. This divided by the horse power will give the rate per horse power per hour.

Cylinder condensation and re-evaporation. When the exhaust valve opens to permit the exit of the steam there

is a perceptible cooling of the walls of the cylinder, especially in condensing engines when a high vacuum is maintained. This results in more or less condensation of the live steam admitted by the opening of the steam valve; but if the exhaust valve is caused to close at the proper time so as to retain a portion of the steam to be compressed by the piston on the return stroke, a considerable portion of the water caused by condensation will be re-evaporated into steam by the heat and consequent rise in pressure caused by compression.

Ordinates. Parallel lines drawn at equal distances apart across the face of the diagram, and perpendicular to the atmospheric line. They serve as a guide to facilitate the measurement of the average forward pressure throughout the stroke, or the pressure at any point of the stroke if desired.

Eccentric. A mechanical device used in place of a crank for converting rotary into reciprocating motion. An eccentric is in fact a form of crank in which the crank pin, corresponding to the eccentric sheave, embraces the shaft, but owing to the great leverage at which the friction between the sheave and the strap acts, compared with its short turning leverage, it can only be used to advantage for the purpose named above.

Eccentric throw is the distance from the center of the eccentric to the center of the shaft. This definition also applies to the term "radius of eccentricity."

Eccentric position. The location of the highest point of the eccentric relative to the center of the crank pin, measured or expressed in degrees.

Angular advance. The distance that the high point of the eccentric is set ahead of a line at right angles with the crank. In other words, the lap angle plus the

lead angle. If a valve had neither lap nor lead, the position of the high point of the eccentric would be on a line at right angles with the crank; as for instance, the crank being at 0° the eccentric would stand at 90° .

Valve travel. The distance covered by the valve in its movement. It equals twice the throw of the eccentric. This refers to engines having a fixed cut off. In the case of an engine with a variable automatic cut off the travel of the cut off valve is regulated by the governor.

Lap. The amount that the ends of the valve project over the edges of the ports when the valve is at mid travel.

Outside or steam lap. The amount that the end of the valve overlaps or projects over the outside edge of the steam port.

Inside lap. The lap of the inside or exhaust edge of the valve over the inside edge of the port.

Lead. The amount that the port is open when the crank is on the dead center. The object of giving a valve lead is to supply a cushion of live steam which, in conjunction with that already confined in the clearance space by compression, shall serve to bring the moving parts of the engine to rest quietly at the end of the stroke, and also quicken the action of the piston in beginning the return stroke.

Compression. Closing of the exhaust passage before the steam is entirely exhausted from the cylinder. A certain quantity of steam is thus compressed into the clearance space.

Throttling governor. Used to regulate the speed of engines having a fixed cut off. The governor controls the position of a valve in the steam pipe, opening or

closing it according as the engine needs more or less steam in order to maintain a regular speed.

Automatic or variable cut off. In engines of this type the full boiler pressure is constantly in the valve chest and the speed of the engine is regulated by the governor controlling the point of cut off, causing it to take place earlier or later according as the load on the engine is lighter or heavier.

Fixed cut off. This term is applied to engines in which the point of cut off remains the same regardless of the load, the speed being regulated by a throttling governor as explained above.

Isochronal or shaft governor. This device in which the centrifugal and centripetal forces are utilized, as in the fly ball governor, is generally applied to automatic cut off engines having reciprocating or slide valves. It is attached to the crank shaft and its function is to change the position of the eccentric, which is free to move across the shaft within certain prescribed limits, but is at the same time attached to the governor. The angular advance of the eccentric is thus increased or diminished, in fact is entirely under the control of the governor, and cut off occurs earlier or later according to the demands of the load on the engine.

Adjustable cut off. One in which the point of cut off may be regulated or adjusted by hand by means of a hand wheel and screw attached to the valve stem, the supply of steam being regulated by a throttling governor.

INDICATOR DIAGRAMS.

In the following study of indicator diagrams all the illustrations are reproductions of actual diagrams taken under ordinary working conditions. Figs. 117 and 118 are here introduced in order to define the different points, lines and curves. Fig. 117 was taken from a large vertical engine with the Corliss valve motion.

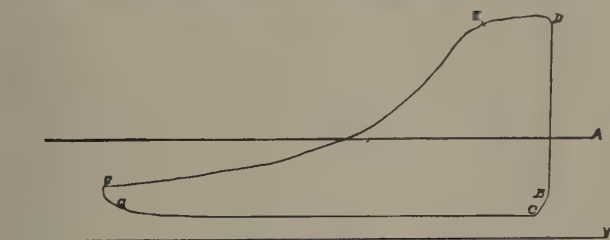


Fig. 117.

The engine being of slow speed and extremely long stroke (10 ft.) with a clearance of but 1 per cent, the compression beginning at C and ending at B is somewhat lighter than is ordinarily given to shorter stroke engines. From B to D is the admission line, which being practically perpendicular to the atmospheric line A, shows sufficient lead and ample port area. From D to E is the steam line. Cut off occurs at E, and from E to F is the expansion curve. At F the point of release is quite sharply defined, as it should be. From F to G is the exhaust line, and from G to C the line of back pressure, sometimes called the line of counter pressure for

In a well proportioned engine having a steam pipe of sufficiently large area, the steam line should parallel the atmospheric line up to the point of cut off. Fig. 118 indicates proper release of the steam at F, and the back pressure from G to C, which is 3 lbs. above the atmospheric line, shows a reasonably free passage of the exhaust steam.

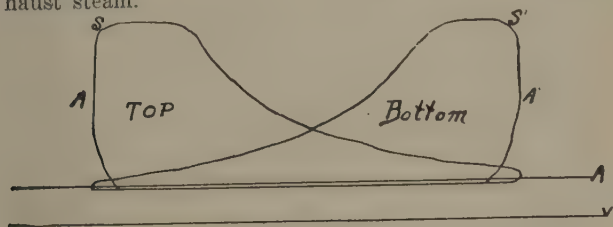


FIG. 119.

Figs. 119 to 124 illustrate diagrams from three new vertical Corliss engines supplying power for an electric lighting plant, which the author was requested to test

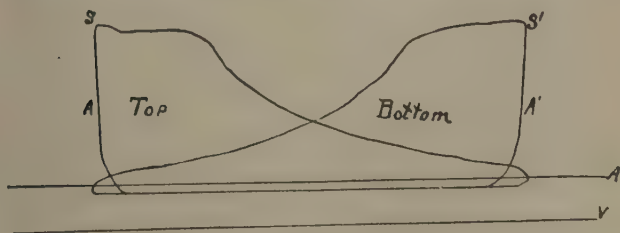


FIG. 120.

and adjust after they had been in operation a few months. The valves had previously been set by the erecting engineer at the time the engines were set up. Each one of these engines exhausted into a separate condenser

of the Jet type, into which the condensing water was forced under pressure and from which the overflow was discharged by gravity into a sewer. There was no air pump and as a consequence the vacuum maintained was very low, usually from 10 to 15 in., and at times still less, so that the beneficial results of condensing were only partially realized.

For convenience the diagrams from each engine will be treated in numerical order, beginning with engine No. 1. This engine was 24 x 48 in., running 70 R. P. M., with a boiler pressure of 68 lbs. A 40 spring was used

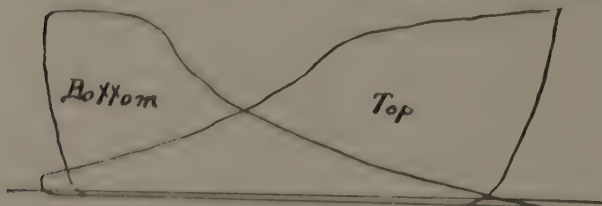


Fig. 121.

in the indicator. The principal defect was the lack of sufficient lead on both ends, as indicated by the inclination inward of the admission lines and the rounded corners of the steam lines at the beginning of the stroke. (See Fig. 119.) There was also more compression, especially on the bottom end, than was necessary, considering the size of the engine and the speed. The necessary changes having been made, the indicator was again applied and the diagram Fig. 120 was obtained, which shows the distribution of the steam to be satisfactory, although at the time of taking this diagram the boiler pressure was only 60 lbs., while it should have been 68

or 70 lbs., because with the latter pressure still better results could have been obtained. The I. H. P. was 235 and the steam used per I. H. P. per hour was 18 lbs.

Fig. 121 is the original diagram from engine No. 2, and shows bad valve adjustment all around, with the exception of lead on the top end. The variation in the points of cut off is the worst feature; cut off taking place on the bottom at 29 per cent. of the stroke, while on the top end it does not occur until the piston has traveled through 42 per cent. of the stroke. There is more

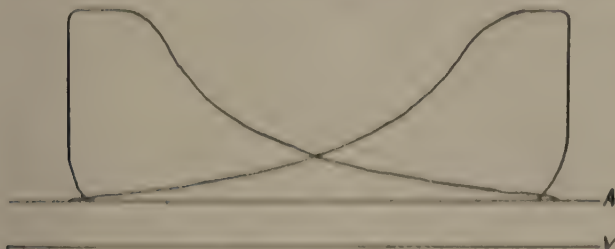


Fig. 122.

compression also than is needed. This engine was 18 x 42 in., running at a speed of 78 R. P. M., and the steam consumption, according to diagram Fig. 121, was 33 lbs. per I. H. P. per hour. Having equalized the cut off and reduced the compression by making the necessary changes in the valve gear, the indicator was again applied, resulting in diagram Fig. 122, which may be considered practically perfect. The boiler pressure was 68 lbs. and the spring used was a No. 40. The steam consumption was reduced to 22 lbs. per I. H. P. per hour as compared to 33 lbs. in Fig. 121.

Figs. 123-124 represent diagrams from engine No.

3, which was the same size as No. 1, viz., 24 x 48 in., and running at 72 R. P. M. The original diagram, Fig. 123, shows too little lead on both ends, but especially on the top. There is also lack of compression on the bottom end. The boiler pressure was 60 lbs. and the scale of

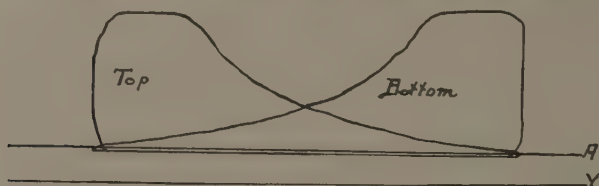


Fig. 123.

spring 40. Fig. 124 taken after the necessary adjustments had been made, shows much better valve performance. The horse power developed was 251 and the steam consumption was 20.5 lbs. per I. H. P. per hour. The rather high rate of steam consumption for this engine

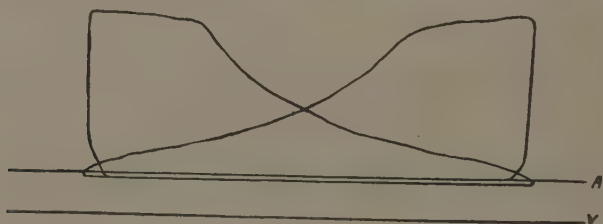


Fig. 124.

as compared with engine No. 1, which was the same size but consumed only 18 lbs. of steam per I. H. P. per hour, was due to two causes. First, a low vacuum; second, low initial pressure necessitating a late cut off.

Figs. 125 to 128, inclusive, are diagrams from a Bullock horizontal non-condensing Corliss engine which had been running about eight or nine months when it fell to the author's lot to apply the indicator to the engine, not only for the purpose of adjusting the valve motion, but also to make a series of tests for the purpose of ascertaining the amount of power delivered by the engine to each one of several different departments which were receiving power from this source.

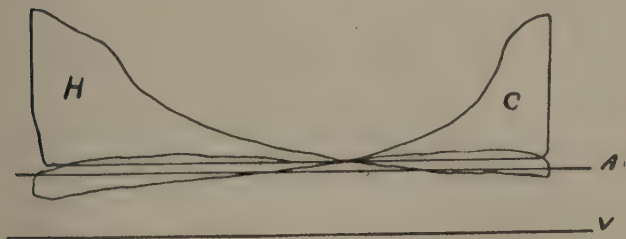


Fig. 125.

The dimensions of the engine were as follows: bore of cylinder, 32 in.; stroke, 5 ft. At the time Fig. 125 was taken the engine was making 62 R. P. M. and the boiler pressure was only 50 lbs. A 30 spring was used. Although the load on the engine was very light at the time, yet the diagram served as a guide to some extent in setting the valves, and by taking off the bonnets from the valve chests and making the necessary changes in the adjustment by the marks on the valves a pretty fair job was made of it, as will be seen by referring to Fig. 126. The reducing motion was a pantograph, and as it is very easy to vary the travel of the paper drum with this motion, diagrams of different lengths were taken

until the one which appeared to be the most satisfactory was obtained. The slight hump in the expansion curve immediately after cut off was probably caused by a speck of dirt or grit which momentarily checked the

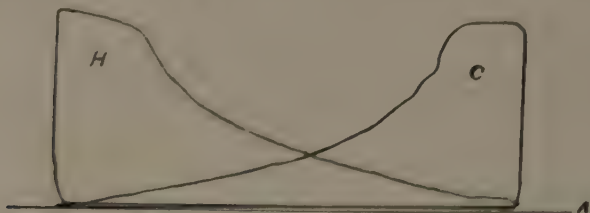


Fig. 126.

indicator piston on the down stroke. The compression on the crank end is not sufficient and the exhaust valve rod on that end was slightly lengthened, resulting in the production of diagram Fig. 127. In this diagram the familiar hump in the crank end expansion curve re-

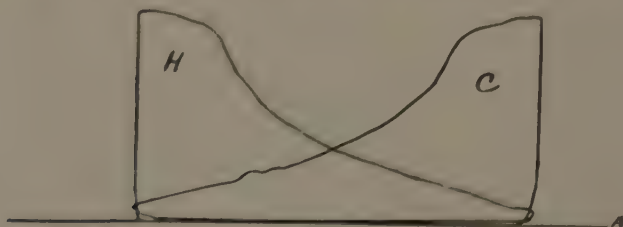


Fig. 127.

appears, but in a different location, being nearer the end of the stroke. It will also be noticed that the length of Fig. 127 has been considerably reduced from that of Figs. 125 and 126, it being about one inch shorter.

The boiler pressure and the load on this engine were gradually increased from time to time, from 50 lbs. and a light load (as shown by Fig. 125), to 60 lbs. and 335 horse power (as indicated by Fig. 126 taken some three months later), and when Fig. 128 was taken, about two years and eight months later, the boiler pressure had been increased to 87 lbs. and the I. H. P. was over 700.

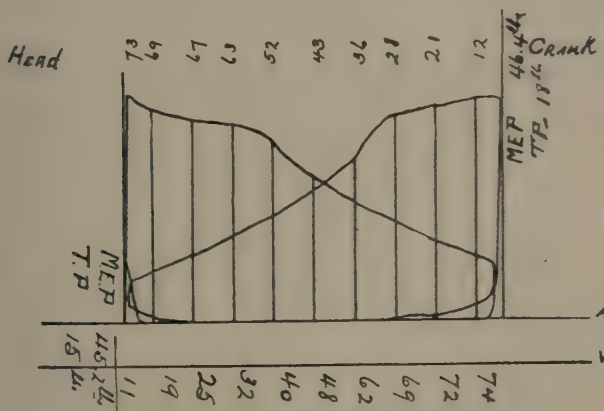


Fig. 128.

Diagram Fig. 128 shows good economy in the use of steam in spite of the fact that the cut off occurs rather late. There is no back pressure worth mentioning, the back pressure line forming part of the atmospheric line through the largest part of the stroke. The reason for this is that the areas of the exhaust ports as well as the exhaust pipe were sufficiently large to permit a free passage for the steam. The exhaust pipe, also, was made as short and direct as possible and all superfluous elbows were dispensed with. The steam con-

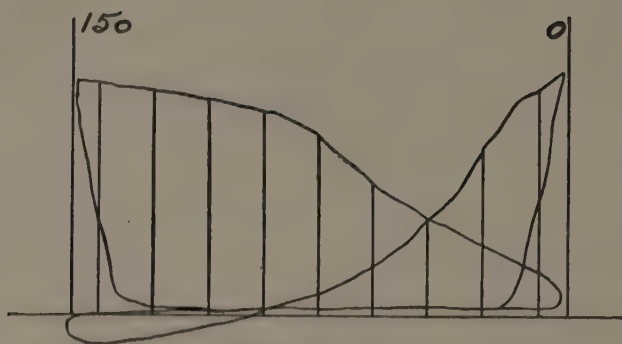


Fig. 120.

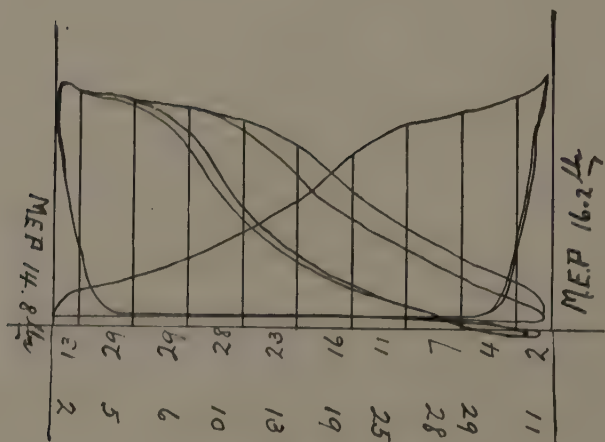


Fig. 130.

sumed per I. H. P. per hour as per diagram Fig. 128 was 22.3 lbs., and the horse power developed was 710.6.

Figs. 129 and 131, inclusive, represent diagrams from a Buckeye engine 24×48 in., and are introduced for the purpose of emphasizing the need of caution and good judgment in setting valves by the indicator when the load on the engine is variable. Fig. 129, which was the first to be taken, would seem to indicate that the valve was badly adjusted, but when Fig. 130 was taken imme-

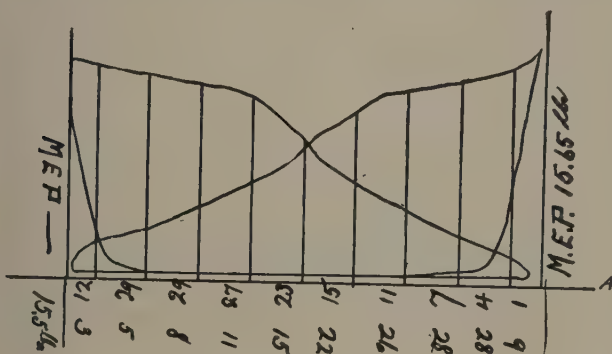


Fig. 131.

diately afterwards, the cause of the trouble became apparent. The engine was furnishing power for operating an electric street railway on a small scale, and the variation in the points of cut off was caused by the stopping and starting of the cars.

Fig. 130 is a notable example of the quick and delicate action of the shaft governor, as it will be seen that during four successive revolutions there was a different load each time, as shown by the diagram from the crank end.

Fig. 131 was secured by quick manipulation of the instrument when it was known that the load was to be steady for a few seconds.

Fig. 132 is from an Atlas single valve automatic cut off engine with shaft governor. This engine was 16×24 in., running at 105 R. P. M., and at the time the diagram was taken the boiler pressure was only 50 lbs. The spring used was a No. 30. The diagram is a fairly good one for the type of engine. Owing to the variation in the angular advance of the single eccentric actuated

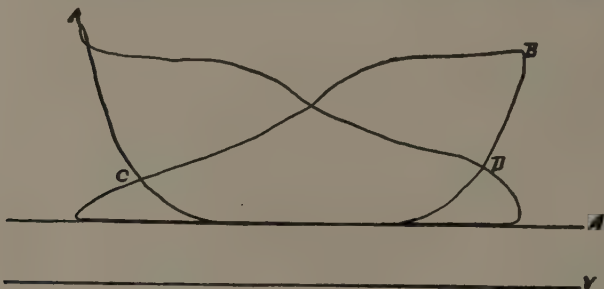


Fig. 132.

by a shaft governor, the degree of compression varies with the point of cut off in the single valve engine, the compression being higher with an early cut off than it is when cut off occurs later in the stroke. The loop at A is caused by too much lead which, together with the compression, caused a momentary rise in the pressure above the normal. The lead at B is approximately correct. The difference in terminal pressures at C and D is the result of shifting of the points of cut off caused by variations in the load. The back pressure lines are almost identical with the atmospheric line, showing that

the exhaust is in no way restricted or cramped. I. H. P. is 65.7 and steam consumption 21 lbs. per I. H. P. per hour.

Figs. 133 and 134 are diagrams taken from a cross compound condensing Corliss engine. The high pressure cylinder was 24×48 in., and the low pressure cylinder was 44×48 in. The steam from the high pressure exhausted into a receiver and from thence into the low pressure cylinder. The receiver pressure was 5.3 lbs. above atmospheric pressure. The ratio of piston areas was 3.36

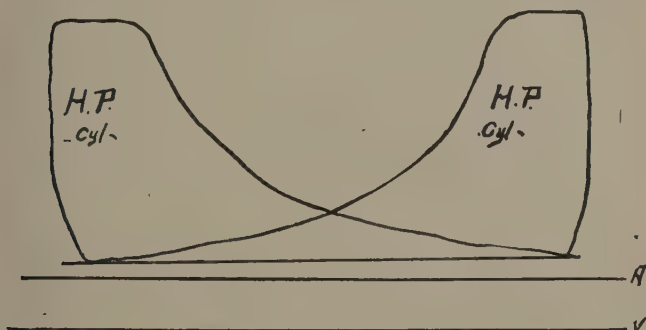


Fig. 133.

to 1. That is, the area of the low pressure piston was 3.36 times the area of the high pressure piston, which was about the correct ratio for the pressure carried, viz., 84 lbs. gauge or 99 lbs. absolute. A No. 40 spring was used on the high pressure and a No. 12 on the low pressure cylinder. The number of expansions in the two cylinders was 14. Thus, the ratio of expansion in the high pressure cylinder was 4.5 and in the low pressure the ratio was 3.1. Then $4.5 \times 3.1 = 14$; or, Thus, initial

pressure=99 lbs. absolute, terminal pressure in L. P. cylinder=7 lbs. absolute; then $99 \div 7 = 14$.

To illustrate the process of finding the M. E. P. without the use of ordinates when the absolute initial and terminal pressures and the number of expansions in each cylinder are known, the following problems will be worked out:

Find M. E. P. in L. P. cylinder.

First, find initial pressure.

Rule. T. P. multiplied by number of expansions. Thus, $7 \times 3.1 = 21.7$ lbs. absolute initial pressure in L. P. cylinder.

Second, find mean forward pressure (M. F. P.).

Rule. Multiply initial pressure by hyperbolic logarithm of number of expansions plus 1, and divide product by number of expansions. Thus the hyperbolic logarithm of 3.1=1.1314, to which add 1=2.1314. Then
$$\frac{21.7 \times 2.1314}{3.1} = 14.9 \text{ lbs. M. F. P.}$$
 Deduct from this the

back pressure, which was 5 lbs. absolute. Thus, $14.9 - 5 = 9.9$ lbs. M. E. P. in L. P. cylinder.

Next find M. E. P. in H. P. cylinder.

First, find T. P. in H. P. cylinder.

This will equal the initial pressure in the L. P. cylinder + 2 per cent for loss in the receiver. Thus, $21.7 + .4 = 22.1$ lbs., terminal pressure in H. P. cylinder.

Second, find initial pressure in H. P. cylinder.

Rule. Multiply T. P. by number of expansions. Thus, $22.1 \times 4.5 = 99.4$ lbs., absolute initial pressure in H. P. cylinder.

Third, find mean forward pressure (M. F. P.).

The hyperbolic logarithm of $4.5=1.5041$, add $1=2.5041$. Then $\frac{99.4 \times 2.5041}{4.5}=55$ lbs., M. F. P. in H. P. cylinder. Deduct back pressure 22.1; thus, 55 lbs.—22.1 lbs.=32.9 lbs., M. E. P. in H. P. cylinder.

The ratio of piston areas being 3.36 to 1, it may be of interest to pursue the subject a little farther and ascertain how the distribution of the steam in the two cylinders corresponds to the ratio of areas. The ratio and pressures may be expressed as follows:

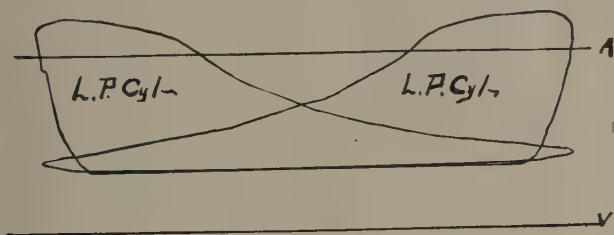


Fig. 134.

Ratio of areas—H. P. cylinder, 1; L. P. cylinder, 3.36. M. E. P.—H. P. cylinder, 32.9; L. P. cylinder, 9.9 lbs. which is very nearly correct; sufficiently so for all practical purposes, and clearly demonstrates that with the intelligent use of the indicator it is possible to so adjust the valves and establish the points of cut off on a compound or triple expansion engine that the work done in each cylinder will be practically the same. As for instance, the product of the area of the H. P. piston and the M. E. P.=14,883.6 lbs., and that of the L. P. piston $\times$ M. E. P.=15,052.9 lbs., a difference of only 169.3 lbs. If the two products had been equal, the horse power exerted in the two cylinders would have been the

same. As it was, the horse power of the H. P. cylinder was 263.4 and that of the L. P. cylinder was 266.4, showing a difference of only three horse power in the amount of work done in each cylinder.

Fig. 135 was taken from one of a pair of Fishkill Corliss engines connected to a common crank shaft. The engines were each 24×48 in., and run at 65 R. P. M., with a boiler pressure of 65 lbs. They were equipped with a jet condenser and a bucket plunger air pump served

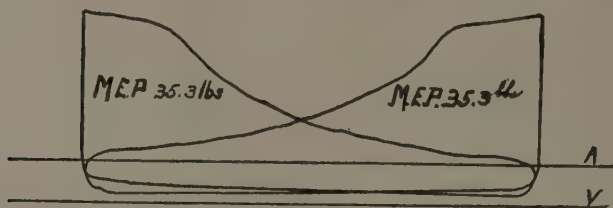


Fig. 135.

for both engines. These engines had been in continuous service for nearly seventeen years when the author was called upon to indicate them and adjust the valves. A diagram taken at the same time from the mate of this engine was very nearly an exact counterpart of Fig. 135. The horse power, as shown by Fig. 135, was 248, and the steam per I. H. P. per hour was 15.2 lbs. The vacuum gauge showed 27 in. and a 50 spring was used.

Figs. 136 and 137 are from an old Fishkill Corliss engine 16×42 in., to which the author applied the indicator after he had set the valves, according to the ordinary rules for valve setting, by the marks placed on the ends of the valves and valve chests. These diagrams are introduced especially for the purpose of showing the

need of exercising the greatest of care to prevent dirt or grit of any kind from getting into the indicator cylinder. After the indicator pipes had been blown out sufficiently, as it was thought, the indicator, which was a thoroughly reliable instrument, was attached and diagram Fig. 136



Fig. 136.

was obtained. It showed the valve adjustment to be very nearly correct, but the perfectly straight steam lines and the sharp corners and sudden drop at cut off were a puzzle, especially in an old engine where the

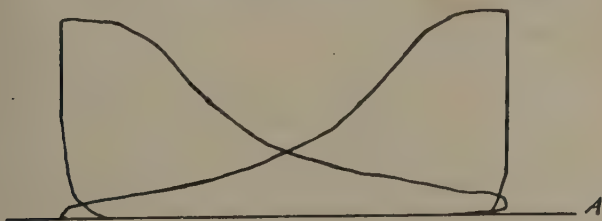


Fig. 137.

valves and valve seats were known to be much worn down. After taking several more diagrams with precisely the same result, the indicator was removed, and upon taking out the piston a quantity of dirt was found on it and also on the inside of the cylinder. This fully

explained the cause of the sharp corners, etc., on the diagram. After the indicator had been cleaned and oiled it was again connected and Fig. 137 was produced, which is a truthful presentation of the performance of the steam in the cylinder.

Many diagrams are misleading, owing to causes similar to the above, and a diagram with too sharp angles at cut off or release should be regarded with suspicion until it is proved beyond all doubt to be truthful.

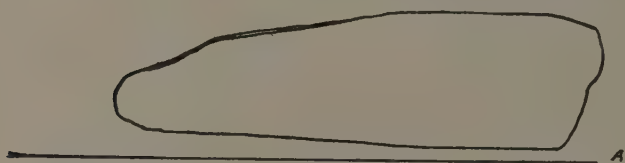


Fig. 138.

Fig. 138 represents a diagram from a vertical non-condensing engine 14×16 in. with riding cut off, which the author was called upon to adjust. This engine was nearly new, having been run but a few months, and although the size of it was ample to do all the work required, yet it had failed, so far, to supply one-half the power needed. After taking the diagram and making a few outside investigations, the cause of the trouble was apparent. Indeed, the wonder was that the engine had supplied as much power as it had under the circumstances.

First. It was situated too far from the boiler plant, being fully 1,200 ft., and although a pressure of 85 lbs. was carried at the boilers and the steam was conveyed through a 6-inch pipe, yet owing to the many drains on the pipe for heating buildings, running other small en-

gines, etc., by the time the steam reached the engine in question the pressure was reduced so much that a 30 spring was found to be too strong, although that was the scale of Fig. 138.

Second. The end of the exhaust pipe was found to be submerged in a nearby pond of water to which it had been carried, probably with a view of making a condensing engine out of it! It was also found that there was no less than four superfluous elbows in the exhaust pipe that could easily be dispensed with. The diagram shows that the cut off was practically useless. That the back pressure was nearly 6 lbs. above the atmosphere, and that the engine was using 55 lbs. of steam and 7 lbs. of coal per horse power per hour, all of which conditions were about as bad as they could be.



Fig. 139.

After increasing the lead and adjusting the cut off a No. 16 spring was used and Fig. 139 was produced which, although still showing late admission, is an improvement over the original diagram. The initial pressure being only 30 lbs. above the atmosphere, further work with the indicator was deferred until changes were made in the steam and exhaust pipes, by which the initial pressure was increased to 55 lbs. and the exhaust pipe

was freed of extra turns and raised from its watery grave into the open air. The engine has since then given perfect satisfaction.

Fig. 140 is from a Buckeye automatic cut off engine 18×36 in. The engine had been running for several years with the valves in the condition shown by the diagram, and in the meanwhile, the load having been increased from time to time, the engine finally refused to run up to speed and something had to be done. The superintendent of the plant said that he had an idea that something was the matter with the engine but could not ascertain what it was, and so he finally called upon the author to apply the indicator. The result was that diagram Fig. 140 was obtained, showing that the principal cause of the trouble was unequal cut off. After

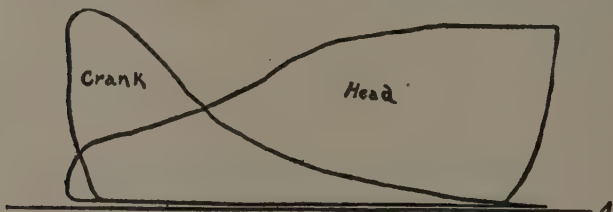


Fig. 140.

equalizing the cut off and increasing the lead on the crank end by a small fraction diagram Fig. 141 was taken, and after this the engine gave no further trouble. The depression in the steam lines might have been rectified to some extent by increasing the boiler pressure, thus giving a higher initial pressure and an earlier cut off. The speed of the engine was 94 R. P. M., with a boiler pressure of 70 lbs. A 40 spring was used with the indicator.

In order to more fully illustrate the process of ascertaining the M. E. P. without dividing the diagram into ordinates, the following computation is given together with rules, etc. In this process two important factors are necessary, viz., the absolute initial pressure and the

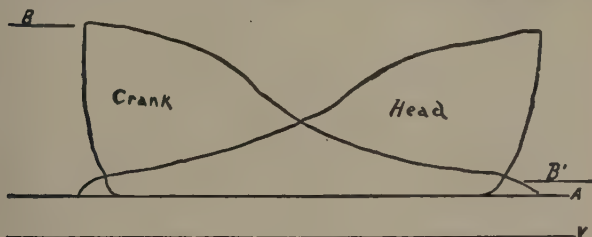


Fig. 141.

absolute terminal pressure, and they can both be obtained from the diagram by measuring with the scale adapted to the spring used. Thus, in Fig. 141 the absolute initial pressure measured from the line of perfect vacuum V to line B is 77 lbs., and the absolute terminal pressure measured from V to line B' is 21 lbs. The ratio, or number of expansions, is found thus:

Rule. Divide the absolute initial pressure by the absolute terminal pressure; thus, $77 \div 21 = 3.65 =$ number of expansions.

Second. Find mean forward pressure.

Rule. Multiply absolute initial pressure by the hyperbolic logarithm of number of expansions plus 1, and divide product by number of expansions. Thus, referring to Table 27, it will be seen that the hyperbolic logarithm of 3.65 is 1.2947, to which 1 must be added.

Then $\frac{77 \times 2.2947}{3.65} = 48.4$ lbs., which is the absolute mean

forward pressure. From this deduct the absolute back pressure, which is 16 lbs., or 1 lb. above atmosphere; thus, $48.4 - 16 = 32.4$ lbs. M. E. P.

Third. Find I. H. P.

Area of piston minus one-half area of rod $\times$ M. E. P. $\times$ piston speed in feet per minute, divided by 33,000. Thus

$$\text{(the diameter of rod being 3 in.)}, \frac{250.96 \times 32.4 \times 564}{33000} = 138.9 \text{ I. H. P.}$$

The steam consumption per I. H. P. per hour may also be computed by means of Table 28, which was originally calculated by Mr. Thomson, and is based upon the following theory:

TABLE 28.

T. P.	W.	T. P.	W.	T. P.	W.
3	117.80	18	466.57	28	798.10
3.5	135.75	18.5	483.43	28.5	814.39
4	153.88	19	500.22	29	830.64
4.5	171.94	19.5	517.07	29.5	846.96
5	186.75	20	533.85	30	863.25
5.5	207.60	20.5	550.64	30.5	879.49
6	225.24	21	567.36	31	895.70
6.5	242.97	21.5	584.10	31.5	911.86
7	260.54	22	600.78	32	927.99
7.5	278.06	22.5	617.40	32.5	944.07
8	295.44	23	633.96	33	960.12
8.5	312.80	23.5	650.46	33.5	976.27
9	330.03	24	666.90	34	992.38
9.5	347.27	24.5	683.38	34.5	1008.46
10	364.40	25	699.80	35	1024.50
10.5	381.57	25.5	716.27	35.5	1040.51
11	398.64	26	732.69	36	1056.48
11.5	415.73	26.5	749.06	36.5	1072.42
12	432.72	27	765.38	37	1088.32
12.5	449.69	27.5	781.76	37.5	1104.35

A horse power=33,000 ft. lbs. per minute, or 1,980,000 ft. lbs. per hour, or $1,980,000 \times 12 = 23,760,000$ in. lbs. per hour, meaning that the same amount of energy required to lift 33,000 lbs. one foot high in one minute of time would lift 23,760,000 lbs. one inch high in one minute of time. Now if an engine were driven by a fluid that weighed one pound per cubic inch, and the mean effective pressure of this fluid upon the piston was one pound per square inch, it would require 23,760,000 lbs. of the fluid per horse power per hour. But if in place of the heavier fluid we substitute pure distilled

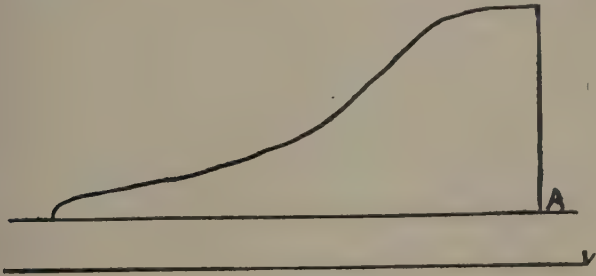


Fig. 142.

water of which it requires 27.648 cu. in. to weigh one pound, the consumption per I. H. P. per hour will be considerably less; as, for instance, $23,760,000 \div 27.648 = 859,375$ lbs., which would be the rate per hour of the water driven engine if the M. E. P. of the water was one pound per square inch and if the M. E. P. was increased to 20 lbs.; then twenty times more power would be developed with the same volume of water, but the weight of water consumed per H. P. per hour would be proportionately less. Now if the engine is driven by steam it will consume just as much less water in pro-

portion as the water required to make the steam is less in volume than the steam used. Therefore if the above constant number, 859.375, be divided by the M. E. P. of any diagram and by the volume of the terminal pressure, the quotient will be the water (or steam) consumption per I. H. P. per hour.

Referring to Table 28, the numbers in the W columns are the quotients obtained by dividing the constant, 859.375, by the volumes of the absolute pressures given in the columns under T. P. and which represent terminal pressures. The table is considerably abridged from the original, which was very full and complete, the pressures advancing by tenths of a pound from 3 lbs. to 60 lbs.; but it is seldom that in ordinary practice there is needed such accuracy. If at any time, however, a diagram should show a terminal pressure not given in the table, the correct factor for that pressure can be easily found by dividing the constant 859.375, by the relative volume of the pressure as found in Table 10 of the properties of saturated steam.

Referring again to Fig. 141, it is seen that the terminal pressure is 21 lbs. absolute, and by reference to Table 28 and glancing down column T. P. until 21 is reached, it will be seen that the number opposite in column W is 732.69. This number divided by the M. E. P. of the diagram Fig. 141, which is 32.4 lbs., gives 22.6 lbs. per I. H. P. per hour as the steam consumption. The rate thus found makes no allowance for clearance and compression, however, and these two very important items will be treated in a succeeding section together with the method of correction for the above, viz., clearance and compression, as they enter largely into the steam economy of an engine.

Steam Consumption from Indicator Diagrams. In calculating the steam consumption of an engine, two very important factors must not be lost sight of, viz., clearance and compression. Especially is this the case in regard to clearance when there is little or no compression, for the reason that the steam required to fill the clearance space at each stroke of the engine is practically wasted, and all of it passes into the atmosphere or the condenser, as the case may be, without having done any useful work except to merely fill the space devoted to clearance. On the other hand, if the exhaust valve is closed before the piston completes the return stroke, the steam then remaining in the cylinder will be compressed into the clearance space and can be deducted from the total volume which, without compression, would have been exhausted at the terminal pressure.

Figs. 142 and 143, which are reproductions of diagrams taken by the author while adjusting the valves on a 16×42 in. Corliss engine, will serve to graphically illustrate this point. Fig. 142, which was the first one to be taken, shows no compression. The point of admission at A is plainly defined by the square corner at the extreme end of the stroke. The clearance of this engine is 4 per cent. of the volume of the piston displacement. The engine being 16 in. bore by 42 in. stroke, the piston displacement is found by the following calculation: Area of piston, 201.06 sq. in. × stroke, 42 in. = 8444.52 cu. in. The volume of clearance space is equal to 8444.52 cu. in. × .04 = 337.78 cu. in., which divided by 1,728 = .195 cu. ft.

By reference to Fig. 143, taken after adjusting the valves for compression, it will be noticed that the steam is there compressed to 37 lbs., the compression curve

beginning at C and ending at B. There is therefore compressed during each stroke a volume of steam equal to .195 cu. ft. at a pressure of 37 lbs. gauge, or 52 lbs. absolute.

One cubic foot of steam at 52 lbs. absolute pressure weighs .1243 lbs., and .195 cu. ft. will weigh $.1243 \times .195 = .0242$ lbs.

The engine was running at 70 R. P. M., or 140 strokes per minute. Thus, according to Fig. 143, the total weight of steam compressed and doing useful work during one hour, and which without compression would have passed out through the exhaust pipe, is equal to $.0242 \times 140 \times 60 = 203.28$ lbs.

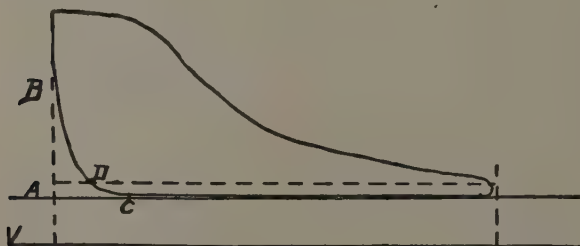


Fig. 143.

Now in order to estimate the steam consumption of the above engine from diagram Fig. 142, it would be necessary to account for all the steam occupying not only the volume of the piston displacement at the end of the stroke, but the clearance as well, for the reason, as before stated, that it would all be released before exhaust closure. This would equal $8444.52 \text{ cu. in.} + 337.78 \text{ cu. in.} = 8782.3 \text{ cu. in.}$, which divided by $1,728 = 5.08 \text{ cu. ft.}$ each stroke, or 10.16 cu. ft. each revolution.

The absolute terminal pressure of Fig. 142 is 20 lbs. One cubic foot of steam at this pressure weighs .0507 lbs., and the weight of steam consumed each revolution would therefore be $10.16 \times .0507 = .515$ lbs., which multiplied by 70 R. P. M. = 36.05 lbs. per minute, or 2,163 lbs. per hour. The horse power developed by the engine at the time was 80. Therefore the steam consumption per I. H. P. per hour = $2,163 \div 80 = 27$ lbs.

Referring again to Fig. 143 it will be remembered that the total weight of steam compressed during one hour was 203.28 lbs. The weight of steam consumed per hour, therefore, equals $2,163 - 203.28 = 1,959.7$ lbs.

Owing to compression, the work area of Fig. 143 is somewhat smaller than that of Fig. 142, amounting in fact to the area of the irregular figure enclosed between the points A, B and C. The work represented by this figure amounts to a very small proportion of the total work indicated by Fig. 142, still in order to arrive at correct conclusions, it should be deducted therefrom.

Assuming the negative work to be equal to .55 horse power, we have $80 - .55 = 79.45$ I. H. P. as the work represented by Fig. 143. As the total weight of steam consumed in one hour was 1,959.7 lbs., the steam consumption per I. H. P. per hour will be $1,959.7 \div 79.45 = 24.67$ lbs., a saving by compression of 2.33 lbs. per H. P. per hour, besides the great advantage of having a cushion of steam in contact with the piston at the termination of the stroke, thus bringing the moving parts of the engine to rest quietly without shock or jar.

The steam consumption may also be computed from the diagram, regardless of the dimensions of the cylinder or the horse power developed. The mean effective pressure and also the absolute terminal pressure must, how-

ever, be known. This method was referred to in the preceding section, but in the computation therein made no correction was made for clearance and compression.

Having reviewed these two factors at considerable length it will now be in order to more fully explain the methods of treating diagrams when it is desired to make these corrections.

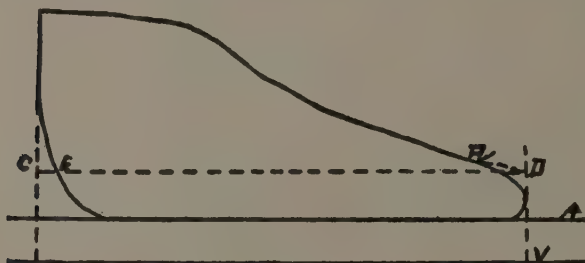


Fig. 144.

First, draw vertical lines C and D, Fig. 144, at each end of the diagram, and perpendicular to the atmospheric line. Draw line V, representing perfect vacuum, 14.7 lbs. below the atmospheric line, as indicated on the scale adapted to the diagram, which in this case is 50 lbs. to the inch. Continue the expansion from R, where release begins, until it intersects line D V, from which point the absolute terminal pressure can be measured.

Having ascertained the terminal pressure, which for Fig. 144 is 30 lbs., draw line D E, which may be called the consumption line for 30 lbs. The terminal being 30 lbs., refer to Table 28 and find in column W, opposite 30 in column T. P., the number 1,024.5. Divide this number by the M. E. P. which in Fig. 144 is 41 lbs., and the quotient, which is 24.99 lbs., is the uncorrected rate of

steam consumption. This rate stands for the total consumption throughout the whole stroke represented on the diagram by the distance from D to C, which measures 3.25 in., but it is evident that there is a small portion of the return stroke, that indicated by the distance from E to C, during which the steam compressed in the clearance space should not be charged to the consumption rate, but should be deducted therefrom. In order to do this, multiply the uncorrected rate by the distance from D to E, which is $3\frac{1}{8}$ in., or 3.125 in., and divide the product by the distance from D to C, $3\frac{1}{4}$ in., or 3.25 in. Thus, $24.99 \times 3.125 \div 3.25 = 24.03$ lbs., which is the corrected rate and represents a saving by compression of $24.99 - 24.03 = .96$ lbs., or nearly 3.7 per cent.

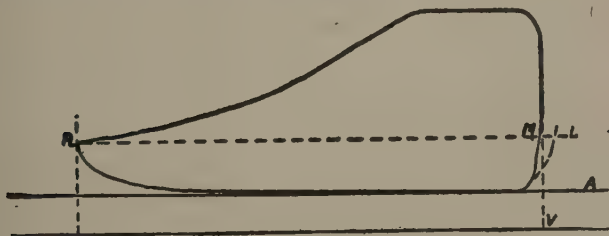


Fig. 145.

In many cases the terminal pressure greatly exceeds the compression, an illustration of which is given in Fig. 145 which is a reproduction of a diagram from an old Wheelock engine. It now becomes necessary to extend the compression curve to L, a point equidistant from the vacuum line with the terminal at R. The consumption line R. L. now becomes longer than the stroke line R. M., therefore the corrected rate will exceed the uncorrected rate by just so much; as for instance, terminal

pressure—34 lbs. The factor, as per Table 28=1152.26, and the M. E. P. of the diagram is 47 lbs. Then, $1,152.26 : 47 = 24.5$ lbs., uncorrected rate; 24.5×3.125 in. (distance R. L.) $\div 3$ in. (distance R. M.) = 25.52 lbs., corrected rate, a loss of a little more than one pound, or about 4 per cent.

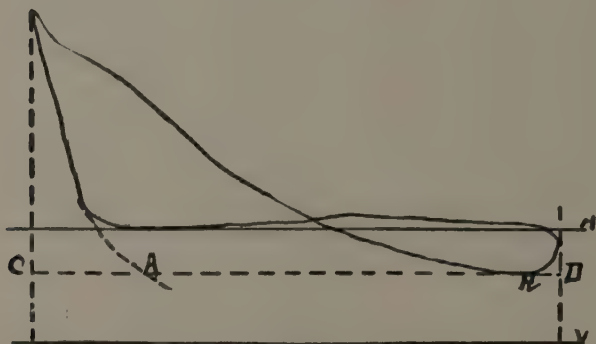


Fig. 146.

There is another class of diagrams very frequently encountered in which the terminal pressure is considerably below the compression curve, and in order to compute the consumption rate by the above method it becomes necessary to continue the compression curve downwards until it meets the terminal, as illustrated at A, Fig. 146, which is a friction diagram from a Buckeye engine. R is the point of release, D A represents the consumption line, and D C the stroke. The terminal is 8.5 lbs., and the factor for that pressure, according to Table 28, is 312.8. Dividing this number by the M. E. P., which was 7 lbs., gives 44.6 lbs. as the uncorrected rate. The distance D to A, where the compression

curve intersects the consumption line, is 2.625 in., and the total length of the diagram C to D is 3.375 in. Then $44.6 \times 2.625 \div 3.375 = 35$ lbs. as the corrected rate. The extremely high rate is owing to the fact that the engine was running light, no load except a line of empty shafting.

Theoretical Clearance. The expansion and compression curves of a diagram are created by the expansion and compression of all the steam admitted during the stroke. This includes the steam in the clearance space as well as in the cylinder proper. It is evident, therefore, that the volume of the clearance is one of the factors controlling the form of these curves, and when the clearance is known a correct expansion or isothermal curve may be theoretically constructed, as will be explained later on. Also if the actual curves, either expansion or compression, of a diagram assume an approximately correct form, the clearance, if not already known, may be determined theoretically from them; although too much confidence should not be put in the results as they are liable to show either too little or too much clearance, generally the latter, especially if figured from the compression curve.

For the benefit of those who may desire to test this method of ascertaining the percentage of clearance of their engines, several illustrations will be given of its application to actual diagrams taken from engines in which the clearance was known.

Fig. 147 is from an engine in which the clearance was known to be 5 per cent. As compression cuts but a very small figure in this diagram, the expansion curve alone will be utilized for obtaining the theoretical clearance, and the process is as follows:

Select two points, C and R, in the curve as far apart as possible, but be sure that they are each within the limits of the true curve. Thus C is located just after cut off takes place, and R is at a point just before release begins. From C draw line C D parallel with the atmospheric line. From D draw line D R, and from C draw line C E, both perpendicular to the atmospheric line. Then from R draw line R E, forming a rectangular parallelogram, C D R E, with two opposite corners,

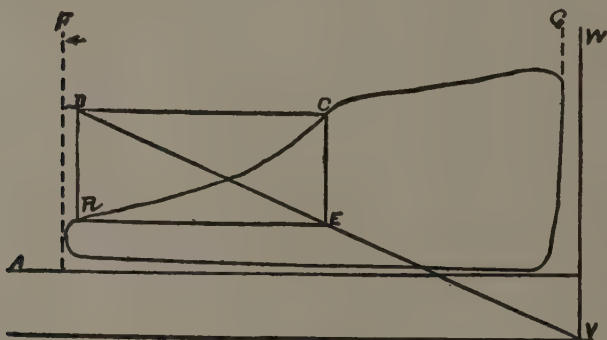


Fig. 147.

C and R, within the curve. Now through the other two corners, D and E, draw the diagonal D E, extending it downwards until it intersects the vacuum line V. From this point erect the vertical line V W, which is the theoretical clearance line.

To prove the result proceed as follows: Measure the length of diagram from F to G, which in this case is 3.75 in., representing piston displacement. Next measure the distance from F to the clearance line V W, which is 3.91 in., representing piston displacement with volume

of clearance added. Then $3.91 - 3.75 = .16$, which represents volume of clearance; and $.16 \times 100 \div 3.75 = 4.3$ per cent., which is approximately near the actual clearance, which, as before stated, was 5 per cent.

Fig. 148 serves to illustrate the same method applied to the compression curve. This diagram is a reproduction of one taken from the low pressure cylinder of a large compound condensing Corliss engine in which the actual clearance was 2.25 per cent. Two points, G and

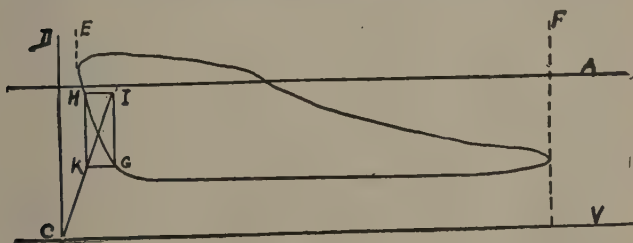


Fig. 148.

H, are selected in the compression curve, and from them the parallelogram G H I K is erected with two of its opposite corners, G and H, well within the limits of the curve, while through the other two corners, I and K, the diagonal I K C is drawn intersecting the vacuum line at C, thus locating the point from which the clearance line C D can be drawn. The measurements in this case are as follows:

Total length of diagram, E to F = 3.75 in.

Distance from clearance line, D C, to F = 3.875 in.

Volume of clearance = $3.875 - 3.75 = .125$ in.

$.125 \times 100 \div 3.75 = 3.33$ per cent. clearance, which is 1.08 per cent. more than the known clearance.

However, notwithstanding the liability to error in

many cases, still this method of computing clearance may often be utilized to good advantage.

Another and more practical method of measuring clearance is as follows: Place the engine on the dead center. Remove the valve chest cover and take out the valve. Close the cylinder cock on that end of the cylinder to which the piston has been moved, leaving the cock on the opposite end of the cylinder open and disconnected from its drip pipe, so as to give an opportunity for catching any water that may leak past the piston while measuring the clearance space. Then having first provided a known weight of water, always making sure of having a little more than enough, pour it into the steam port until the clearance space is filled to a level with the valve seat. When this is done, weigh the water that is left and deduct it from the original quantity, and the remainder will be the number of pounds of water required to fill the clearance, from which it is an easy matter to compute the number of cubic inches or cubic feet in the space devoted to clearance. If any water leaks past the piston during the operation it should be weighed and deducted from the total quantity poured into the port.

In the case of an engine having the valve chest on the side of the cylinder it will be necessary to close the steam port either by blocking the valve against it or by fitting a piece of soft wood into it, making it water tight. The water can then be poured into the clearance space through a pipe connected to the indicator opening in that end of the cylinder. Care should be exercised to allow a vent for the air to escape as it is displaced by the water.

The Theoretical Expansion Curve. According to

Boyle's law the volume of all elastic gases is inversely as their pressures, and steam being a gas conforms substantially to this law; although the expansion curves of indicator diagrams are affected more or less by the loss of heat transmitted through the cylinder walls, and by the change in the temperature of the steam produced by the changes in pressure during the progress of the stroke. The pressure generally falls more rapidly during the first part of the stroke, and less rapidly during the last portion than it should in order to conform strictly to the above law, and the terminal pressure usually is greater than it should be to agree with the ratio of expansion. But this fullness of the expansion curve of the diagram near the end compensates in a measure for the too rapid fall near the beginning of the stroke. Therefore, if the engine is in fairly good condition with the valves properly adjusted and not leaking, and the piston rings are steam tight, it may be assumed that the expansion of the steam in the cylinder takes place according to Boyle's law and it is found that the expansion curve drawn by the indicator practically coincides with a hyperbolic curve constructed according to that law.

Fig. 149 graphically illustrates the application of the hyperbolic law to the expansion of gases. The horizontal lines represent volumes and the vertical lines represent pressures. The base line, A F, represents the full stroke of a piston in the cylinder of an engine, and the vertical line, A I, represents the pressure of the steam at the commencement of the stroke.

Suppose there is no clearance and that the steam has been admitted up to point II when it is cut off. The rectangle A B II I is the product of the pressure mul-

tiplied by the volume of the steam thus admitted. When the piston has traveled from A to C the volume of the steam has been doubled and the pressure C L has been reduced to just one-half what it was at A I, but the area of the rectangle A C L M is equal to the area of the initial rectangle, and, as before, is the product of the pressure C L multiplied by the volume A C. As the piston travels still farther, as from A to D, the steam is expanded to four volumes while the pressure at D K will only be one-fourth that of the initial pressure; but the new rectangle A D K N is still equal in area to either of the others, A B H I or A C L M.

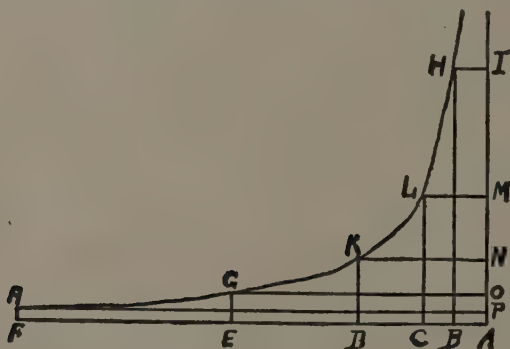


Fig. 149.

The same law applies to each of the remaining rectangles; A E G O representing five volumes and one-fifth of the initial pressure, and A F R P representing six times the initial volume and one-sixth of the initial pressure, but each having the same area as the initial rectangle A B H I. Now the area of the rectangle A B H I represents the work done by the steam up to the point of

cut off, and the area of the hyperbolic figure enclosed by the lines $B H R F$ represents the work done by the expansion of the steam after cut off occurs. This area and the amount of work it represents may be computed by means of the known relations of hyperbolic surfaces with their base lines; as for instance, if the base lines $A B$, $A C$, $A D$, etc., extend in geometrical ratio, as 1, 2, 4, 8, 16, etc., the successive areas, $B H L C$, $B H K D$, $B H G E$, etc., increase in arithmetical ratio, as 1, 2, 3, 4, etc.

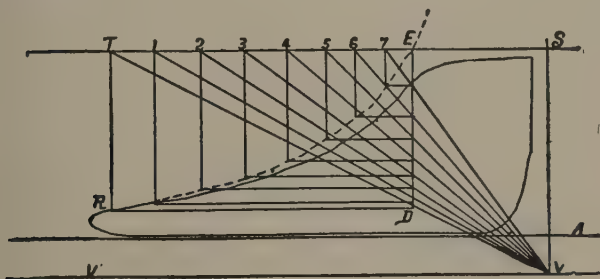


Fig. 150.

On the principles of common logarithms, which represent in arithmetical ratio natural numbers in geometrical ratio, tables of hyperbolic logarithms have been computed for the purpose of facilitating the calculation of areas of work due to different degrees of expansion.

A theoretical curve may be constructed conjointly with the actual expansion curve of a diagram by first locating the clearance and vacuum lines and then pursuing the method illustrated by Fig. 150. A curve so produced is called an isothermal curve, meaning a curve of the same temperature.

Referring to Fig. 150, suppose, first, that it is desired to ascertain how near the expansion curve of the diagram coincides with the isothermal curve, at or near the point of cut off. Select point R near where release begins, but still well within the expansion curve. From this point draw the vertical line, R T, parallel with the clearance line, V S. Then draw the horizontal line, S T, parallel with the atmospheric line, and at such a height above it as will equal the boiler pressure as measured by the scale adapted to the diagram; such measurement to be made from the atmospheric line to correspond with the gauge pressure. From T draw the diagonal T V, and from R draw the horizontal line R D parallel with the atmospheric line. From D, where this line intersects T V, erect the perpendicular D E, thus forming the parallelogram R D E T, and as line T V passes through two of its opposite angles and meets the junction of the clearance and vacuum lines, the other two angles, R and E, will be in the theoretical curve, and R being the starting point, it is obvious that this curve must pass through E, which would be the theoretical point of cut off on the steam line S T.

Two important points in the theoretical curve have now been located, viz., E as the cut off, and R as the point of release. In order to obtain intermediate points, draw any desired number of lines downward from points in S T, as 1, 2, 3, 4, 5, etc., and continue them downwards far enough to be sure that they will meet the intended curve, and from the same points in S T draw diagonals 1 V, 2 V, 3 V, 4 V, 5 V, etc., all to converge accurately at V. From the intersection of these diagonals with D E draw horizontal lines parallel with V V', and the points of junction of these lines with the vertical

lines will be points in the theoretical curve. It will now be an easy matter to trace the curve through these points. If, on the other hand, it be desired to compare the curves toward the exhaust end of the diagram, draw lines $E D$ and $E T$, Fig. 151, also $T R$, locating R near where release commences, after which draw line $R D$, completing the parallelogram $E T R D$, fixing R as a point in the theoretical curve started at E . After drawing the diagonal $T V$, proceed in the same manner as before to locate the intermediate points.

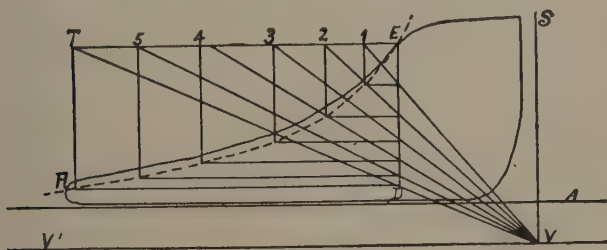


Fig. 151.

It will be observed that in order to ascertain the performance of the steam near the beginning of the stroke, the starting point of the isothermal curve must be near the point of release, and conversely, if the starting point of the curve is located near the point of cut off and coincident with the actual curve, the test will apply towards the end of the stroke. It is not to be expected that the expansion curve of any diagram taken in practice will conform strictly to the lines of the isothermal curve, especially towards the latter end of the stroke, owing to the reëvaporation of water resulting from the condensation of steam which was retained in the cylinder

by the closing of the exhaust valve. This reëvaporation commences just as soon as the temperature of the steam, owing to the reduction of pressure due to expansion, falls below the temperature of the cylinder walls, and it continues at an increasing rate until release occurs. The tendency of this reëvaporation or generation of steam within the cylinder during the latter portion of the stroke is to raise the terminal pressure considerably above what it would be if true isothermal expansion took place. The terminal pressure may also be augmented by a leaky steam valve, while, on the other hand, a leaky piston would cause a lowering of the terminal and an increase in the back pressure.

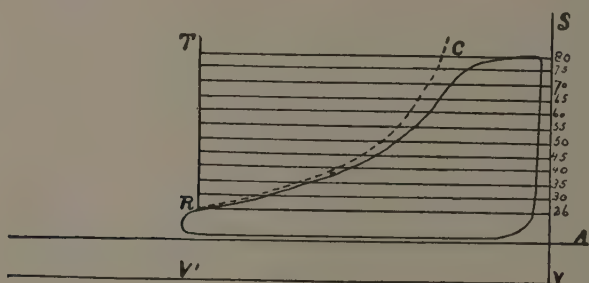


Fig. 152.

The Adiabatic Curve. If it were possible to so protect or insulate the cylinder of a steam engine that there would be absolutely no transmission of heat either to or from the steam during expansion, a true adiabatic curve or "curve of no transmission" might be obtained. The closer the actual expansion curve of a diagram conforms to such a curve, the higher will be the efficiency of the engine as a machine for converting heat into work.

Fig. 152 illustrates a method of figuring a curve which, while not strictly adiabatic, will be near enough for all practical purposes, while at the same time it will give the student an opportunity to study the laws governing the expansion of saturated steam.

To draw the curve, first locate the clearance and vacuum lines $V S$ and $V V'$. Next locate point R in the expansion curve near where release begins, making this the starting point, and also the point of coincidence of the expansion curve with the adiabatic curve. The other points in the curve are located from the volumes of steam at different pressures during expansion; the pressures being measured from the line of perfect vacuum, and the volumes from the clearance line.

The absolute pressure at R , Fig. 152, is 26 lbs. From point R erect the perpendicular $R T$. Also draw horizontal line $R 26$ parallel with the vacuum line and at a height equal to 26 lbs. above vacuum line $V V'$, as shown by the scale, which in this case was 40. The length of line $R 26$, measured from R to the clearance line, is $3 \frac{1}{16}$ in., or 3.0625 in. By reference to Table 10 it will be seen that the volume of steam at 26 lbs. absolute, as compared with water at 39° , is 962. Now if the length of line $R 26$ be divided by this volume, and the quotient multiplied by each of the volumes of the other pressures represented at points 30, 35, 40, 45, etc., up to the initial pressure, the products will be the respective distances from the clearance line of points in the adiabatic curve. These points can be marked on the horizontal lines drawn from the clearance line to line $R T$.

Starting with line $R 26$, it has been noted that its length is 3.0625 in., and that the volume was 962. $3.0625 \div 962 = .003$. Then the volume of steam at 30 lbs. is

841, which being multiplied by $.003=2.5$ in., the length of line 30. Next the volume at 35 lbs. $=728$. Multiplying this volume by $.003=2.1$ in., length of line 35, and so in like manner for each of the other points.

The process involves considerable figuring and careful and accurate measurements, which should be made with a steel rule with decimal graduations. It is not expected that the cut Fig. 152 will be found accurate enough in its measurements to serve as a standard; it being intended only to serve as an illustration of the process. The diagram from which the illustration was drawn was taken from a 600 H. P. engine situated some 200 ft. from the boilers, and there was a considerable cooling of the steam by the time it reached the engine, the effect of which is apparent. The curve produced by the measurements is shown by the broken line. The process can be applied to any diagram.

Power Calculations. The area of the piston (minus one-half the area of rod) multiplied by the M. E. P., as shown by the diagram, and this product multiplied by the number of feet traveled by the piston per minute (piston speed) will give the number of foot pounds of work done by the engine each minute, and if this product be divided by 33,000, the quotient will be the indicated horse power (I. H. P.) developed by the engine.

Therefore one of the first requisites in power calculations is to ascertain the M. E. P. Beginning with the most simple, though only approximately correct, method of obtaining the average pressure, as illustrated by Fig. 153, draw line A B touching at A and cutting the diagram in such manner that the space D above it will equal in area spaces C and E taken together, as nearly as can be estimated by the eye. Then with the scale meas-

ure the pressure along the line F G at the middle of the diagram, which will be the M. E. P.

The process is based upon the theory that the average width of any tapering figure is its width at the middle of its length. This method should not be relied upon as accurate, but is convenient at times when it is desired to make a rough estimate of the horse power of an engine.

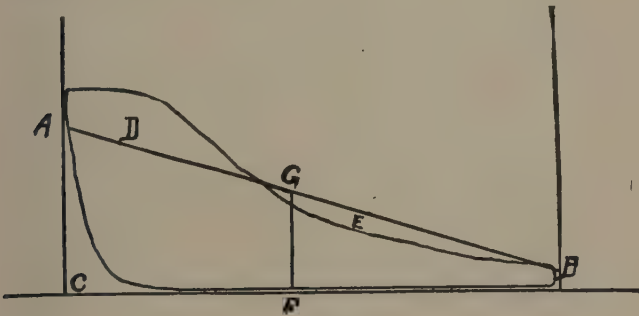


Fig. 153.

Figuring the M. E. P. by Ordinates. This is a very common method and one which can be relied upon to give accurate results, provided care is exercised in its use.

The process consists in drawing any convenient number of vertical lines perpendicular to the atmospheric line across the face of the diagram, spacing them equally, with the exception of the two end spaces, which should be one-half the width of the others for the reason that the ordinates stand for the centers of equal spaces, as for instance, line 1, Fig. 154, stands for that portion of the diagram from the end to the middle of the space

between it and line 2. Again, line 2 stands for the remaining half of the second space and the first half of the third, and so on. This is an important matter, and should be thoroughly understood, because if the spaces are all made of equal width, and measurements are taken on the ordinates, the results will be incorrect, especially in the case of high initial pressure and early cut off, following which the steam undergoes great changes.

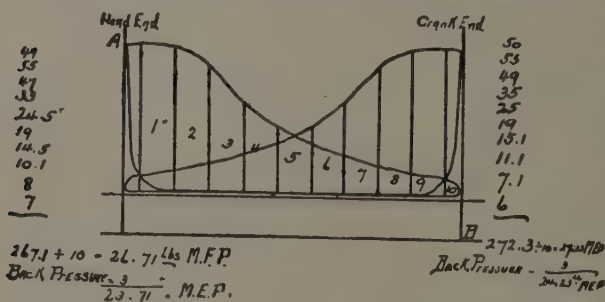


Fig. 154.

If the spaces are all made equal, the measurements will require to be taken in the middle of them, and errors are liable to occur, whereas if spaced as before described, the measurements can be made on the ordinates, which is much more convenient and will insure correct results. Any number of ordinates can be drawn, but ten is the most convenient and is amply sufficient, except in case the diagram is excessively long. For spacing the ordinates, dividers may be used, or a parallel ruler may be procured from the makers of the indicator; but one of the most convenient and easily procurable instruments for this purpose is a common two-foot rule, and the method of using it is illustrated in Fig. 154.

First draw vertical lines at each end of the diagram, perpendicular to the atmospheric line and extending downwards to the vacuum line, or below it if necessary, in order to have a point on which to lay the rule. In Fig. 154 points A and B are found to be the most convenient. Now lay the rule diagonally across the diagram, touching at A and B, and the distance will be found to be $3\frac{3}{4}$ in., or 60 sixteenths.

Suppose it be desired to draw 10 ordinates. Divide 60 by 10, which will give 6 sixteenths, or $\frac{3}{8}$ in. as the width of the spaces, but as the two end spaces are to be one-half the width of the others, there will be 11 spaces altogether, the two outer ones having a width equal to one-half of $\frac{3}{8}$ or $\frac{3}{16}$. Now apply the rule again in the same manner, touching at points A and B, and with a sharp pointed pencil begin at A and mark the location of the first ordinate according to the rule, at a distance of $\frac{3}{16}$ from the end. Then $\frac{3}{8}$ from this mark make another one, which will locate the second ordinate, and proceed in like manner to locate the others. The last two or three marks generally come below the diagram, and if the diagram be taken from a condensing engine it may be necessary to tack it on to a larger sheet of paper in order to get these points. Having correctly located the ordinates, they may now be drawn perpendicular to the atmospheric line or vacuum line, either of which will answer.

It should be noted that, owing to the diagonal position of the rule with relation to the atmospheric line, the spaces are not of the actual width as described by the rule, but this is unimportant, so long as they are of a uniform width. This method can be applied to any diagram, no matter what its length may be, and point

B may be located at any distance below the atmospheric or vacuum lines, wherever it is the most convenient for the subdivisions on the rule, sixteenths, eighths, etc., so long as it is in line with the end of the diagram. Having thus drawn the ordinates, the M. E. P. may be found by measuring the pressure expressed by each one, using for this purpose the scale adapted to the spring used, adding all together and dividing by the number of ordinates, which will give the average pressure.

Referring to Fig. 154, begin with ordinate No. 1 on the diagram, from the head end of the cylinder. In this case a 40 spring was used. Lay the scale on the ordinate with the zero mark where it intersects the compression curve. The pressure is seen to be 49 lbs. Set this down at that end of the card and measure the pressure along ordinate No. 2, which is 55 lbs. Proceed in this manner to measure all the ordinates, placing the resulting figures in a column, after which add them together and divide by 10. The result is 26.71 lbs., which is the mean forward pressure (M. F. P.). To obtain the mean effective pressure, deduct the back pressure, which is represented by the distance of the exhaust line of the diagram above the atmospheric line in a non-condensing engine, and in a condensing engine the back pressure is measured from the line of perfect vacuum, 14.7 lbs., according to the scale below the atmospheric line.

In Fig. 154 the back pressure is found to be 3 lbs. Therefore the M. E. P. of the head end will be $26.71 - 3 = 23.71$ lbs. On the crank end the M. F. P. is 27.23 lbs., and $27.23 - 3 = 24.23$ lbs. = M. E. P. The average effective pressure on the piston, therefore, will be $23.71 + 24.23 \div 2 = 23.97$ lbs.

Unless great care is exercised in the measurements, errors are liable to occur in applying this method, especially with scales representing high pressures, as 60, 80, etc. The most convenient and reliable method is to take a narrow strip of paper of sufficient length, and starting at one end, apply its edge to each ordinate in succession and mark their lengths on it consecutively with the point of a knife blade or a sharp pencil. Having thus marked on the paper the total length of all the ordinates, ascertain the number of inches and fractions of an inch thereon, the fractions to be expressed decimally, and divide by the number of ordinates. The quotient will be the average height of the diagram, and as the scale expresses the number of pounds pressure for each inch or fraction of an inch in height, if the average height of the diagram be multiplied by the number of the scale, the product will be the M. F. P.

Referring again to Fig. 154, if the lengths of the ordinates drawn on the head end diagram be measured, their sum will be found to be $6 \frac{8}{12}$ or 6.666 in. Dividing this by 10 gives .666 in. as the average height. The mean forward pressure will then be as follows: $.666 \times 40 = 26.64$ lbs., or practically the same as found by the other method.

Fig. 155 illustrates a type of diagram frequently met with, and one which requires somewhat different treatment in estimating the power developed. It will be noticed that, owing to light load and early cut off, the expansion curve drops considerably below the atmospheric line, notwithstanding that the engine from which this diagram was taken is a non-condensing engine. When release occurs at R, and the exhaust side of the

piston is exposed to the atmosphere, the pressure immediately rises to a point equal to, or slightly above, that of the atmosphere.

Fig. 155 was taken during a series of experiments made by the author for the purpose of ascertaining the friction of shafting and machinery, and the engine it was obtained from is a Buckeye 24x48 in. The boiler pressure at the time was only 40 lbs., and a No. 20 spring was used. The ordinates are drawn according to the method illustrated in Fig. 154. By placing the

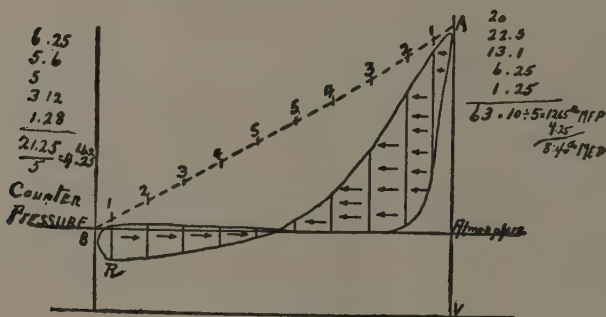
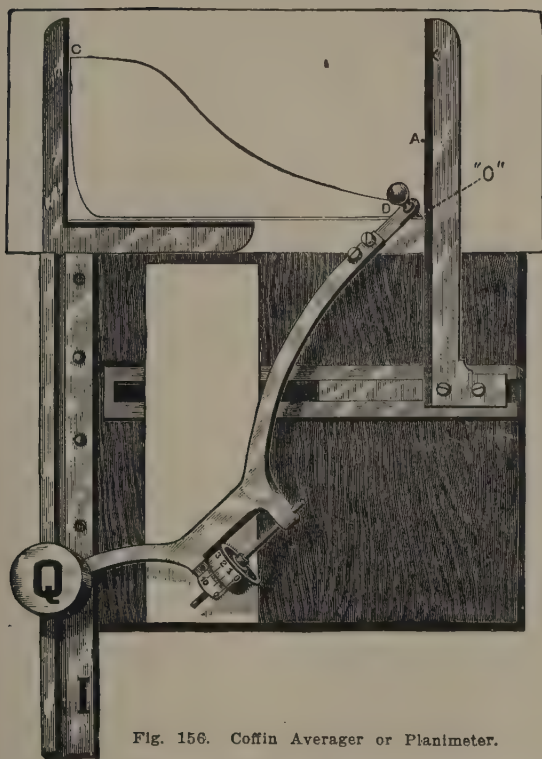


Fig. 155.

rule on points A and B, the distance between those two points is found to be $3\frac{5}{8}$ in., or 58 sixteenths. Dividing this by 10 gives 5.8 sixteenths, or nearly $\frac{3}{8}$ in., as the width of the spaces; the two end spaces being one-half of this, or $\frac{3}{16}$ in. wide. The first five ordinates, counting from A, express forward pressure, represented by the arrows. The remaining five ordinates, counting from B, express counter or back pressure, represented by the arrows pointing in the opposite direction. Measuring the pressures along the first five ordi-



nates, and adding them together, gives 63.1 lbs., which divided by 5, gives 12.65 lbs. as the mean forward pressure (M. F. P.).

Then figuring up the counter pressure in the same manner on the other five ordinates, beginning at B, the result is 4.25 lbs. The M. E. P. therefore will be $12.65 - 4.25 = 8.4$ lbs.

Obtaining the M. E. P. with the Planimeter. The area of the diagram represents the actual work done by the steam acting upon the piston. In a non-condensing engine the lower or exhaust line of the diagram must be either coincident with or slightly above the atmospheric line in order to express positive work. Any deviation of this line, either above or below the atmospheric line, represents counter pressure, the amount of which may be ascertained by measurements with the scale, and should be deducted from the mean forward pressure.

On the other hand, the exhaust line of a diagram from a condensing engine falls more or less below the atmospheric line, according to the degree of vacuum maintained, and the nearer this line approaches the line of perfect vacuum, as drawn by the scale, 14.7 lbs. below the atmospheric line, the less will be the counter pressure, which in this case is expressed by the distance the exhaust line is above that of perfect vacuum.

The prime requisite therefore in making power calculations from indicator diagrams is to obtain the average height or width of the diagram, supposing it were reduced to a plain parallelogram instead of the irregular figure which it is.

The planimeter, Figs. 156 and 157, is an instrument which will accurately measure the area of any plane surface, no matter how irregular the outline or boundary line is, and it is particularly adapted for measuring the areas of indicator diagrams, and in cases where there are many diagrams to work up, it is a very convenient instrument and saves much time and mental effort. In fact, the planimeter has of late years become an almost indispensable adjunct of the indicator.

It shows at once the area of the diagram in square inches and decimal fractions of a square inch, and when the area is thus known it is an easy matter to obtain the average height by simply dividing the area in inches by the length of the diagram in inches. Having ascertained the average height of the diagram in inches or fractions of an inch the mean or average pressure is

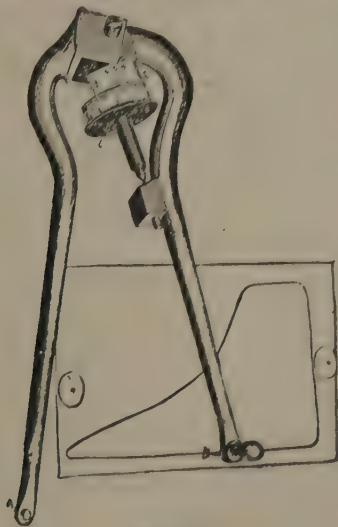


FIG. 157.

found by multiplying the height by the scale. Or the process may be made still more simple by first multiplying the area, as shown by the planimeter in square inches and decimals of an inch, by the scale, and dividing the product by the length of the diagram in inches. The result will be the same as before, and troublesome fractions will be avoided.

QUESTIONS.

170. Into what two general classes are steam engines divided.

Ans. Simple and compound.

171. Describe a simple engine.

Ans. A simple engine may be either condensing or non-condensing, but its leading characteristic is, that the steam is used in but one cylinder.

172. What is a condensing engine?

Ans. One in which the exhaust steam is passed into an air tight vessel in which a vacuum is maintained, the exhaust steam being there condensed by coming in contact with cold water, or a series of tubes through which cold water is being circulated.

173. Describe a compound engine?

Ans. A compound engine is one in which the steam is made to do work in two or more cylinders before it is allowed to exhaust.

174. How is this accomplished?

Ans. By causing the exhaust steam from the first, or high pressure cylinder, to pass into a second cylinder of larger diameter, and, if the engine be triple or quadruple expansion, from thence into a third or fourth cylinder, the diameters of which increase in regular ratio.

175. What is a non-condensing engine?

Ans. One from which the steam exhausts directly into the atmosphere, or is used for heating purposes before passing out into the open air.

176. What disadvantage does a non-condensing engine constantly labor under?

Ans. The pressure of the atmosphere amounting to 14.7 lbs. per sq. in. is constantly in resistance to the motion of the piston.

177. Mention several other causes that tend to increase the back pressure upon the piston of a non-condensing engine.

Ans. The resistance of bends and turns in the exhaust pipe, also causing the exhaust to pass through feed water heaters or heating coils.

178. What is back pressure?

Ans. Pressure that tends to retard the forward stroke of the piston.

179. What advantage has a condensing engine over a non-condensing engine?

Ans. The atmospheric pressure is removed from in front of the piston to a degree corresponding to the height of the vacuum that is maintained in the condenser.

180. How many classes of condensers are there in general use?

Ans. Two; jet condensers and surface condensers.

181. Describe a jet condenser.

Ans. One in which the steam is exhausted into an air-tight vessel, and is there condensed by coming in contact with a jet or spray of cold water.

182. How is this water removed?

Ans. By means of the air pump, which also maintains a vacuum in the condenser.

183. Describe a surface condenser.

Ans. It is an air-tight vessel, either cylindrical or rectangular in shape, fitted with a large number of brass or copper tubes, of small diameter, through which the cold water is forced by the circulating pump. A

vacuum is maintained in the body of the condenser by the air pump, and the steam exhausted into this is condensed by coming in contact with the cool surface of the tubes. In some cases the steam passes through the tubes in place of around them, the condensing water being forced into and through the body of the condenser, and the vacuum being maintained in the tubes.

184. Describe an injector condenser.

Ans. A condenser in which the cold water is forced through an annular enlargement of the exhaust pipe, and passing down into a nozzle which gradually contracts. The exhaust steam entering at the same time is condensed, the water rushing through the nozzle with a velocity sufficient to create a vacuum.

185. About what quantity of water is required per horsepower per hour to condense the exhaust steam from an engine?

Ans. About 38 to 40 gals., depending upon the temperature of the condensing water.

186. What three factors are necessary to insure good economy with multiple cylinder engines?

Ans. First—A high initial pressure.

Second—Expansion of the steam to greatest extent possible.

Third—Protecting the surfaces of the cylinders from cooling influences.

187. Describe a cross compound engine.

Ans. An engine consisting of two cylinders, each having its own connecting rod and crank, the cranks being set at opposite ends of the engine shaft, and at an angle of 90° to each other. The high pressure cylinder exhausts into the low pressure cylinder, usually through a receiver.

188. Describe a tandem compound engine.

Ans. An engine having the two cylinders arranged tandem to each other, with a common piston rod and connecting rod.

189. What advantage is gained by this design?

Ans. A much shorter and more direct route for the exhaust steam in its passage from the high to the low pressure cylinder.

190. What are some of the pointers gained by the use of the indicator?

Ans. First—It shows the average pressure upon the piston throughout the stroke.

Second—It shows the action of the valve or valves in admission, cut off and release of the steam.

191. What is meant by absolute pressure?

Ans. Pressure reckoned from a perfect vacuum. It equals the boiler pressure plus the atmospheric pressure.

192. What is boiler or gauge pressure?

Ans. Pressure above atmospheric pressure.

193. What is initial pressure?

Ans. Pressure in the cylinder at beginning of stroke.

194. What is terminal pressure?

Ans. Pressure in the cylinder at end of the stroke.

195. What is mean effective pressure?

Ans. The average pressure acting upon the piston throughout the stroke.

196. What is ratio of expansion?

Ans. A comparison of the volume of steam in the cylinder at point of release with the volume at point of cut off.

196a. What is condenser pressure?

Ans. Pressure existing in the condenser, due to lack of a perfect vacuum.

197. What is a perfect vacuum?

Ans. That condition existing within a closed vessel from which all matter, including air, has been expelled.

197a. Can the vacuum in the condenser of an engine be considered as power?

Ans. It cannot, yet its existence there increases the capacity of the engine for doing work.

198. What is absolute zero?

Ans. A temperature 461.2° below zero Fahr.

199. What is piston displacement?

Ans. The volume or space swept through by the piston in a single stroke.

200. What is piston clearance?

Ans. The distance between the piston and cylinder head when the piston is at the end of the stroke.

201. What is steam clearance?

Ans. The space between the piston at the end of the stroke and the valve face.

202. What is a horsepower?

Ans. 33,000 lbs. raised one foot high in one minute of time.

203. What is indicated horsepower?

Ans. The horsepower as shown by the indicator diagram.

204. What is piston speed?

Ans. The distance in feet traveled by the piston in one minute of time.

205. Define Boyle's Law.

Ans. The pressure of a gas at a constant temperature varies inversely as the space it occupies.

206. What is an adiabatic curve?

Ans. A curve showing the expansion of a gas which loses no heat while expanding.

207. What is an isothermal curve?

Ans. A curve showing the expansion of a gas having a constant temperature, but partially influenced by moisture.

208. What is an expansion curve?

Ans. The curve traced upon paper by the indicator pencil, showing the actual expansion of the steam in the cylinder.

209. What is the first law of thermodynamics?

Ans. Heat and mechanical energy are mutually convertible.

210. What is power?

Ans. The rate of doing work, or the number of foot pounds exerted in a given time.

211. What is a foot pound?

Ans. The unit of work, or the raising of one pound one foot high.

212. What is the first law of motion?

Ans. All bodies continue either in a state of rest or of uniform motion in a straight line, except in so far as they may be compelled by impressed forces to change that state.

213. What is work?

Ans. The product of the pressure or force exerted upon a body, multiplied by the distance passed through in a given time.

214. What is momentum?

Ans. The product of mass and velocity.

215. What is dynamics?

Ans. The science of moving powers.

216. What is force?

Ans. That which alters the motion of a body, or puts in motion a body that was at rest.

217. What is the mechanical equivalent of heat?

Ans. 778 foot pounds.

218. What is steam efficiency?

Ans. Heat converted into useful work, divided by heat expended.

219. How can maximum efficiency be obtained in a steam engine?

Ans. By using the steam at as high an initial pressure as safety permits, and at as large a ratio of expansion as possible.

220. What is meant by steam consumption of an engine?

Ans. The weight in pounds of steam used in one hour, divided by the horsepower developed.

221. What is an eccentric?

Ans. A mechanical device used in place of a crank for converting rotary into reciprocating motion. Its center of rotation is apart from its center of formation.

222. What is eccentric throw?

Ans. The distance from the center of the shaft to the center of the eccentric.

223. What is eccentric position?

Ans. The location of the highest point of the eccentric relative to the center of the crank pin, expressed in degrees.

224. What is angular advance?

Ans. The distance that the high point of the eccentric is set ahead of a line at right angles with the crank.

225. What is valve travel?

Ans. The distance covered by the valve in its movement. It equals twice the throw of the eccentric.

226. What is lap?

Ans. The amount that the valve projects over the edges of the ports when the valve is at mid travel.

227. What is lead?

Ans. The amount that the port is open when the crank is on the dead center.

228. What is meant by dead center?

Ans. That position of the moving parts of the engine when the center of the crank pin and the center of the piston rod are exactly in line.

229. What is the object of giving a valve lead?

Ans. To supply a cushion of live steam to quicken the action of the piston in beginning the stroke.

230. What is compression?

Ans. Closing of the exhaust passage before the steam is entirely released, thus compressing it in the clearance space.

231. Give the rule for calculating the horsepower developed by an engine.

Ans. Area of piston multiplied by the M. E. P., multiplied by piston speed, divided by 33,000.

232. What is a planimeter?

Ans. An instrument that will accurately measure the area of any plane surface, no matter how irregular it may be.

VALVE ADJUSTMENT.

It goes without saying that every man who aspires to be an engineer should endeavor to thoroughly acquaint himself with the principles governing the action of valves as well as the details of valve adjustment. But it must be remembered that this knowledge cannot be acquired in a day or a week, or even months. True, a man may be able to learn some of the alphabet of valve lore in a comparatively short time, but the more practical experience he has in the work the more will he realize the supreme need of mastering all the details of the process.

The common D slide valve, simple as it appears, is capable of furnishing problems over which savants have puzzled themselves.

The development of the full amount of power of which the engine is capable, its efficiency and economical use of steam and its regular and quiet action are, in the largest degree, dependent upon the correct adjustment of its valve or valves.

There are many different types of valves for controlling the admission and release of steam to and from the cylinders of engines, but the basic principles governing the adjustment of all, whether slide, poppet, rotative, piston, etc., are exemplified in the action of the common D slide valve, viz., the admission of the steam to the cylinder, its cut off and release, and the closure of the exhaust, each and all of which events are to take place at the proper moment during one stroke of the piston.

In order to properly perform these important func-

tions the valve must have lead and lap. The various terms relating to valve action are plainly defined in "Definitions," and it is unnecessary to repeat them here. If the outside lap is increased admission will be later and cut off earlier, and if it be desired to keep the lead the same it will be necessary to move the eccentric forward, which will make the other events, cut off, release and compression, earlier also. If the inside lap is increased the result will be an earlier closing of the exhaust and increased compression.

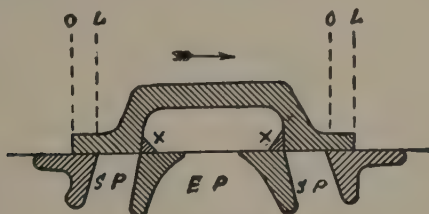


Fig. 158.

These propositions refer mainly to engines of the single valve variety in which one valve controls the admission and distribution of the steam for both ends of the cylinder. In engines of the four-valve type, having a separate steam and exhaust valve for each end of the cylinder, each individual valve may be adjusted independently of the others, as will be explained later on, and in the case of engines having separate eccentrics, one for the steam, and one for the exhaust valves, the adjustment becomes still more perfect.

We will first study the action of the D slide valve by referring to Fig. 158, which is a sectional view of a valve, valve seat and ports. The valve is represented at mid travel or in its central position. S P, S P are

opening, Fig. 160, the crank having moved through 60° and the eccentric is now at 180° , the farthest point of its throw in that direction, the valve being at the end of its travel. At this point it might be well to note a matter about which some persons are liable to become confused, simple as it is, viz., that the travel of a slide valve equals twice the port opening plus twice the outside lap. For instance, suppose the width of each steam port to be $1\frac{1}{4}$ in. and the outside lap to be 1 in. In Fig. 160 the valve is at the extreme end of its travel

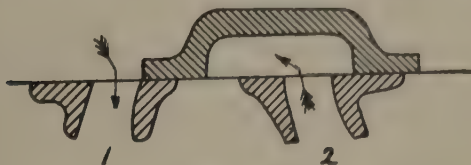


Fig. 160.

towards the right and is about to return. It first covers port number one = $1\frac{1}{4}$ in. Next it moves to mid travel lap number one = $2\frac{1}{4}$ in. Its next move is lap number two = $3\frac{1}{4}$ in., and lastly it uncovers port number two = $4\frac{1}{2}$ in., which is its travel.

To return to the third function of the valve or cut off, Fig. 161. The crank has now traversed 120° and the highest point of the eccentric is at 60° on the return circle, a point equivalent to 240° of the circle described by the crank.

The fourth function is when compression begins at the head end of the cylinder, Fig. 162. The crank is now at 150° , the piston being near the end of the stroke and the eccentric has reached 90° of the return circle or three-quarters of the crank circle, while the crank

has still to travel 30° in order to complete the first one-half of its circle. At this point we can study the effect of inside lap, because if the valve has no inside lap, release on the crank end will begin almost at the same



FIG. 161.

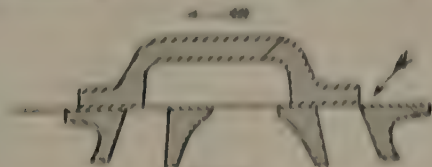
moment that compression takes place at the head end, but by adding inside lap, compression can be caused to take place earlier and release later.



FIG. 162.

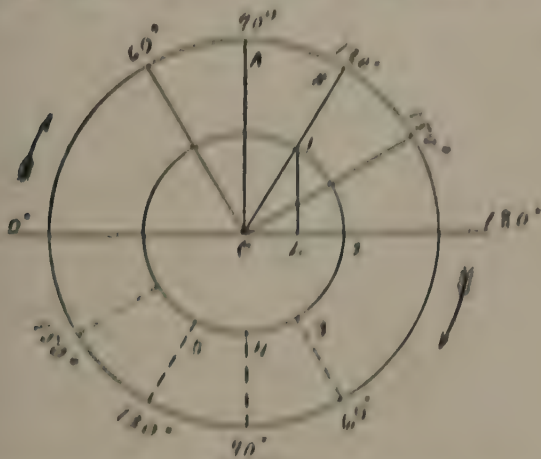
The next event is admission at the head end of the cylinder, Fig. 163. The crank has now arrived at 180° , having completed one-half of a revolution, the piston is at the end of the stroke, and the eccentric is at 120° on the return path. Fig. 164 serves to better illustrate the relative positions of the crank pin and eccentric during the stroke. The inner circle represents the path described by the high point of the eccentric, and the large circle that of the crank pin. The radius $C'P$ of the small circle represents the throw of the eccentric,

and the distance Cl is the lap of the valve plus the lead. The point of intersection of the vertical line, l , with the eccentric circle locates the position of the high-



111 111

est point of the eccentric, and the line $C' B$, drawn from the center of the crank shaft through this point, indicates the angular advance which in this case is 30° .



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represented by the angle $\Delta \theta$. The figures 1, 2, 3, 4, 5 indicate the position of the high point of the eccentric at the moment of each function of the valve. The action

D, 1, 2, 3, 4, 5, 6 equals the radius of eccentricity. Line C D represents the position of the crank when the valve is at mid travel or in its central position, D being the location of the crank pin. Referring back to Fig. 158, the valve is there shown in its central position and supposed to be moving in the direction of the arrow in order to admit steam to the crank end of the cylinder. Again referring to Fig. 165, draw line C A in such a position that the angle A B C will equal the angular advance of the eccentric, which we will assume in this case to be 30° . This will bring the high point of the eccentric at B while the crank, as before stated, is at D. Next using line C A as the diameter, draw a circle about it called the valve circle. Now suppose the crank to be turning in the direction of the arrows. At position D the crank line is just about to cut into the valve circle, the valve being central. When the crank gets to position 1 the valve has moved the distance C E. When the crank is at 2 the valve has moved the distance C M, and when the crank arrives at 3 the valve has moved to the limit of its travel from its central position and it now begins the return movement. The motion of the valve is comparatively slow at this point for the reason that the high point of the eccentric is now passing the center at 7. The distance the valve has moved backward while the crank has moved from 3 to 4 is the distance B F, while F C represents its distance from the central position, and G C the same when the crank is at 5. When the crank arrives at 6 and its line has left the valve circle, the valve is again central. Fig. 165 merely shows the movement of the valve through one-half of its travel without giving any details regarding port openings, cut off, etc.

lap, and the port begins to uncover at this point. Then by the time the crank gets to the center, 1, the port is open the distance L O, which is the lead, in this case $\frac{1}{8}$ in. This position of the valve is shown in Fig. 159.

The position of the crank when cut off takes place is ascertained by drawing a line, C G 5, through the intersection of the outside lap and valve circles, where the valve is on its return movement (see Fig. 161). Thus far no account has been taken of release and compression, and in order to determine the position of the crank when these events occur it will be necessary to draw the valve circle for the opposite movement of the valve, for be it remembered that the movement of the valve so far considered has been only one-half of its travel; that is, it has moved from its central position towards the head end of the cylinder and back again. We have seen how it has thus performed the functions of admission, full port opening and cut off for the crank end of the cylinder, and now by referring to Fig. 167 it will be seen at what point of the stroke the remaining events, viz., release and compression, occur.

Draw a second valve circle, Fig. 167, diametrically opposite the first. Also draw an arc with a radius equal to the inside lap, which in this case is assumed to be one-half inch. When the crank gets to the position 7 its center line cuts the intersection of the inside lap and valve circles, and release begins. When the crank arrives on the center 8, the valve has moved the distance C T from central position; but C X of this distance has been occupied by the inside lap, therefore the lead on the exhaust is represented by the distance X T. When the crank on its return stroke arrives at the position marked 10, its line again cuts the inter-

section of the inside lap and valve circles and compression takes place, as in Fig. 162. By dropping perpendiculars from the positions of the crank at 1, 5, 7 and 10 an indicator diagram may be drawn showing the performance of an engine with this style of valve.

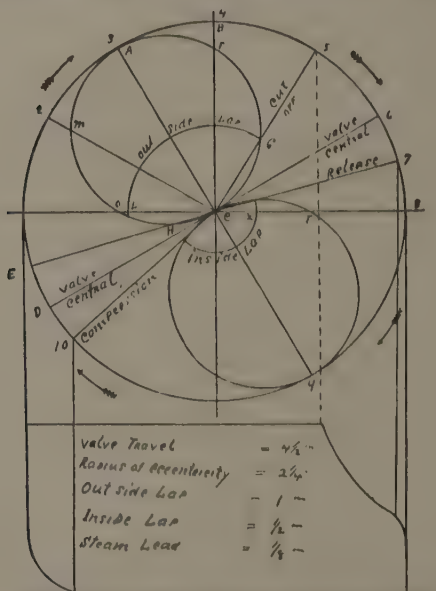


Fig. 167.

Fig. 168 shows the effect of decreasing the angular advance, that is, setting the eccentric back towards the crank. In this instance the eccentric is set back 10°, thus making the angle of advance 20° instead of 30°, as before. The full lines represent the new angle, while

the dotted circles and lines indicate the valve and its movements as drawn at first. A shows the original point of admission and A' the position of the crank when admission takes place with the lesser angle of advance. Similarly, R and R' show the old and new points of re-

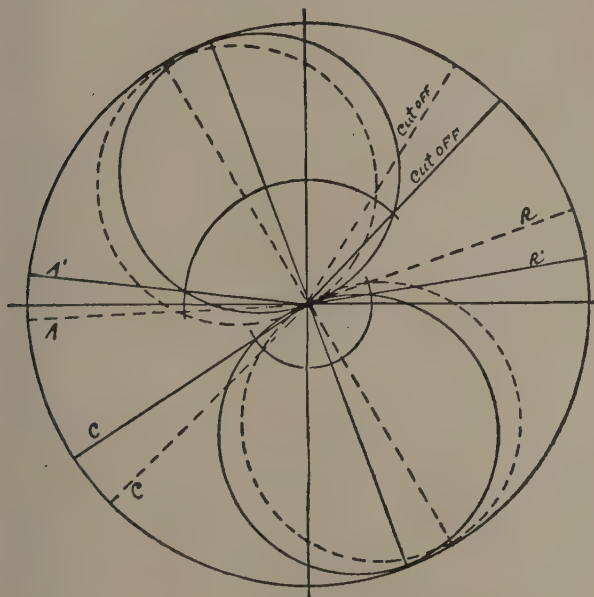


Fig. 168.

lease, and C and C' the compression. The two different points of cut off are also indicated. It will be observed that all of these events occur later and the lead also is diminished.

In locomotives, and also in some types of adjustable

cut off engines, the travel of the valve may be varied at will, and the effect of decreasing the valve's travel is illustrated by Fig. 169, the full lines showing the decreased travel and its influence, and the dotted lines

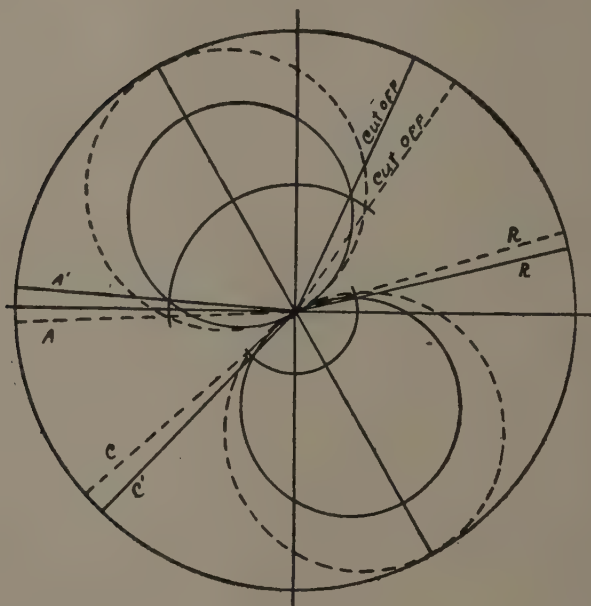


Fig. 169.

showing the original. Admission and release occur later, while cut off and compression take place earlier, and the lead is less. The travel of the valve as indicated in Fig. 169 has been decreased one inch, making it $3\frac{1}{2}$ in. in place of $4\frac{1}{2}$ in. as before.

Fig. 170 shows the result of increasing the outside lap. The lap has been increased in this case from 1 in., as originally drawn, to $1\frac{1}{4}$ in., as indicated by the full lines, while the dotted lines show the lap as it was

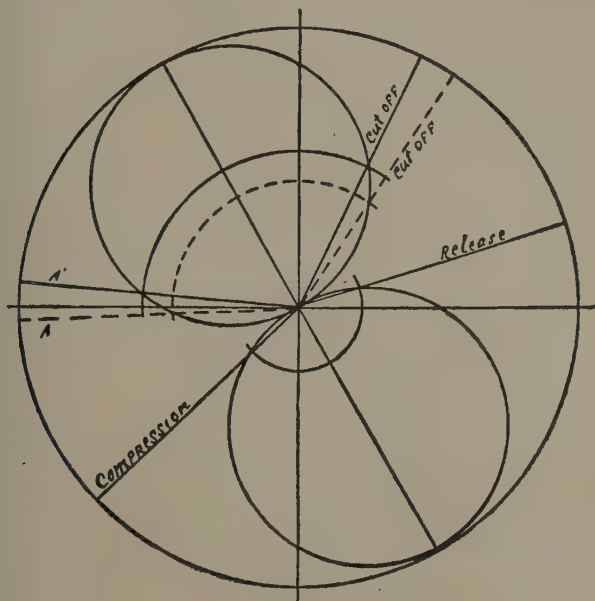


Fig. 170.

before being changed. The effect of this change is to cause less lead, a later admission and an earlier cut off, but compression and release are not affected for the reason that these latter events are controlled by the inside lap, which has not been changed.

In Fig. 167 the valve is shown as cutting off the steam

when the crank has completed 120° , or two-thirds of the half revolution, but the point of cut off on the indicator diagram shows that the piston has traveled $7/9$ of the stroke. This discrepancy is due to the obliquity of the connecting rod, as it will be seen by looking at the valve diagram, Fig. 167, that the crank must travel farther to complete the stroke from this point than the piston does. In order to cause the valve to cut off earlier, say at one-half stroke, it will be necessary to do one of two things, either to increase the outside lap, which would have a tendency to cause admission to occur too late, or the angle of advance may be increased sufficient to cause cut off to take place at half stroke, but to do this alone would cause admission to occur too early. Therefore the proper thing to do is to increase both the angle of advance and the outside lap. Fig. 171 shows how this can be done without decreasing the travel of the valve. The angle of advance, $A B C$, is now 50° , where before it was 30° , as in Fig. 167.

The valve is central when the crank is at position 1; the high point of the eccentric being at point 4. The outside lap, which before was 1 in., has had $7/16$ in. added to it, making it $17/16$ in. When the crank gets to D the port is just commencing to open, and with the crank on the center at 2, the lead is $1/4$ in.

It will readily be seen at this point that by increasing the outside lap still more the lead can be diminished, and the point of cut off made still earlier, but this would result in a still further reduction of the power of the engine, which has already been considerably reduced, as shown by the diminished area of the indicator diagram as compared with the one in Fig. 167. When the crank gets to position 3 the valve has reached the

limit of its travel, and the port is open the distance Aa , which is as far as the outside lap will permit. With the crank at point 4 cut off occurs. But with the increased angular advance and the inside lap remaining

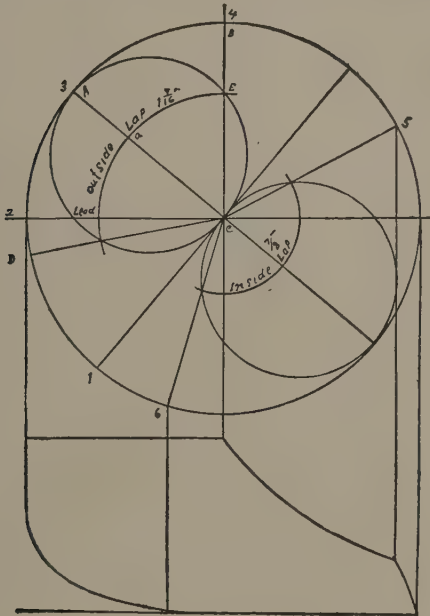


Fig. 171.

as it was before, viz., $\frac{1}{2}$ in., release would occur too early. Therefore it will be necessary to increase the inside lap sufficient to cause release and compression to take place at as near the proper points as possible. In this instance $\frac{3}{8}$ in. has been added, making the inside

lap $\frac{7}{8}$ in., and release takes place with the crank at position 5, while compression begins at 6. These points may also be changed by simply adding to or decreasing the inside lap.

It should be noted that in the foregoing discussion of valve gear it is understood that the valve stem moves in the same direction as the eccentric rod, that is, the direction of motion is not reversed by a rocker arm interposed between the eccentric and the valve.

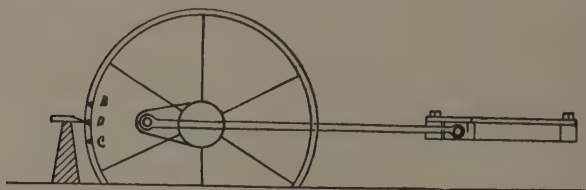


Fig. 172.

The first step in the operation of valve setting is to place the engine on the dead center, which means that the piston is at the end of the stroke, and the centers of the main shaft, crank pin and crosshead pin, or wrist pin, as it is sometimes called, are in line (see Fig. 172). When moving the engine to place it on the center it should always be turned in the direction in which it is to run. This is to guard against any errors which might result from lost motion or looseness in the reciprocating parts. Turn the fly wheel around until the crosshead is almost to the end of the stroke, say within a half inch of it, as at Fig. 173. Then with a steel scriber or penknife mark the location of the crosshead on the guides A, also provide a secure resting place upon the floor of the engine-room for a marker to be

placed against the rim of the wheel. This rest should be firmly fastened to the floor in order that its position may not be changed during the operation of valve setting. Place the marker against the wheel, as at B, and mark the point with a center punch or cold chisel. Next



Fig. 173.

turn the engine carefully until the crosshead completes the stroke and moves back on the return stroke until the mark A is in line again. Make another mark on the rim of the wheel opposite the marker at C. This position of the engine is shown in Fig. 174, and it will be

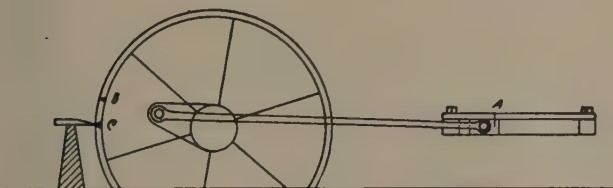


Fig. 174.

seen that the crank is now as much above the center as it was below in Fig. 173. Now with a pair of large dividers ascertain the middle or half distance between marks B and C and put another mark, D, at this point. Then turn the engine a complete revolution until mark

D comes opposite the pointer, Fig. 172, and the engine will be on the true center.

At this point the question may arise, why not simply reverse the motion and back the wheel up until the mark D is in line with the marker? The answer is, that

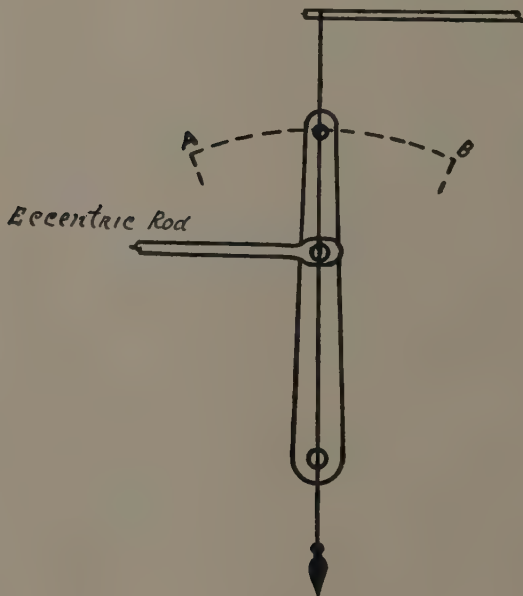


Fig. 175.

while this would undoubtedly save considerable labor, yet it would almost certainly result in an error, on account of the lost motion of the moving parts, which would permit of considerable movement of the wheel before any movement of the crosshead would take place

if the wheel was turned back. The result would be that when mark D came to be opposite to the pointer, the crank would not be on the true center. The next move is to see that the eccentric rod is adjusted to the proper length. If there is a rocker arm, connect the eccentric rod in its proper place, leaving the valve rod disconnected for the time being. Then adjust the length of the rod so that when the eccentric is turned around on the shaft the rocker arm will vibrate equal distances

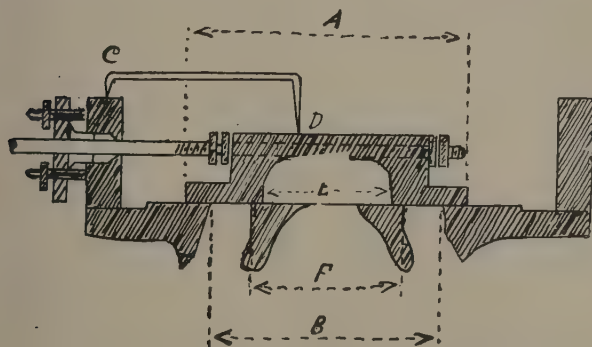


Fig. 176.

on each side of a plummet line suspended through the center of the pin upon which the arm turns, as in Fig. 175. Before connecting the valve rod the valve should be put in its central position and marked. To do this it will be necessary to first ascertain the outside lap.

The most accurate method of doing this is to take the valve out and measure the distances between the outside edges of the steam ports, as at B, Fig. 176. Then measure the width of the valve from edge to edge, as at A. Then $A - B \div 2 =$ the outside lap. For in-

stance, $A = 8.5$ in., $B = 6.5$ in. Then $8.5 - 6.5 = 2$, and 2 divided by 2 $= 1$ in., which is the lap. The inside lap should also be measured at this point for convenience, and the measurements preserved for future reference. The inside lap is ascertained by measuring the distance between the inside edges of the ports and the distance across the arch of the valve from one inside edge to the other (see Fig. 176) and dividing the difference by 2. For instance, the distance F is 4 in., and E is 3 in.; then $\frac{4-3}{2} = .5$ in., making the inside lap $1\frac{1}{2}$ in.

To place the valve central, measure the width of the outside lap each way from the outside edges of the steam ports and mark the points on the valve seat with a sharp lead pencil. Then place the valve with edges on the marks and it will be central. To insure accuracy, measurements should also be taken from the outside edges of the steam ports to the ends of the seats. Having fixed the valve in its central position, replace the stem and if it is secured in the valve by nuts, as in Fig. 176, care should be taken to leave a little play for the valve between the nuts, otherwise it is liable to become stuck and held off the seat when it gets hot and expands. Make a center punch mark C, on the edge of the valve chest directly over the valve stem, and placing one leg of a tram or pair of dividers in the mark, with the other leg describe a mark on the top of the valve as at D, thus marking the valve in its central position.

Now with the rocker arm perpendicular, the eccentric rod having been previously adjusted, connect the valve rod to the rocker, and turn the eccentric to the

limit of its throw in one direction, and measure the distance the valve has traveled from its central position. Then turn the eccentric around to its extreme throw in the other direction, and if the valve travels the same distance from its central position in the opposite direction the lengths of the rods are correct, but if not correct, the necessary change can usually be made by shifting the nuts on the valve stem, or if the valve is secured to the stem by a yoke the change can be made in the rod.

Having succeeded in getting the correct travel for the valve, the next step is to set the eccentric. With the engine on the dead center, turn the eccentric around on the shaft in the direction in which the engine is to run, so as to take up all the play in the valve stem and other moving parts, and with the tram, or dividers watch the valve until it has moved away from its central position by the amount of its outside lap, plus the lead it is desired to give the valve. For instance, if the valve has one inch outside lap and the lead is to be $\frac{1}{8}$ in., the valve should be moved away from its central position $1\frac{1}{8}$ in., and also away from the end of the cylinder at which the piston is. The steam port for that end should now be open $\frac{1}{8}$ in., and the eccentric should be ahead of the crank one-quarter turn plus the angular advance required for the outside lap and lead, or if as previously explained, the motion of the eccentric is reversed by a rocker arm the eccentric should be behind the crank by the same amount. Tighten the set screws holding the eccentric on to the shaft and turn the engine around until it is on the opposite center. Then if the lead is the same on each center the valve is set correctly. If the lead is not the same, move the valve on

the stem toward the end having the most lead, a distance equal to one-half the difference between the two leads. If the lead as equalized is more than is desired move the eccentric back on the shaft until the correct lead opening is secured, then tighten the set screws permanently, and with a sharp cold chisel make a plain mark on the shaft, and opposite to this another mark on the eccentric. This will save considerable trouble in case the eccentric should slip or be accidentally moved from its true position at any time.

Although the common D slide valve as applied to stationary engines usually has its point of cut off fixed, yet there are many types of variable automatic cut off engines with single slide valves of various patterns, such as box valves in which the steam passes through the valve, piston valves, in which the steam either passes through or around the ends of the valve, so-called grid-iron valves, and various other types. Such valves are generally applied to high speed engines, and are actuated by eccentrics which are under the control of shaft governors which vary the position of the eccentric with relation to the crank according to the load that is on the engine, thus regulating the point of cut off so as to maintain a constant speed, while the throttle is kept wide open. While the details of setting all the various styles of valves, including the Corliss or four-valve type, differ considerably from those required in setting the D valve, yet the same principles govern the operation, no matter what kind of a valve is to be adjusted.

In all types of reciprocating engines the same factors affecting the distribution of the steam are present, viz., the outside or steam lap affecting admission and cut off, and the inside or exhaust lap affecting release and

compression. While the D valve (and other types of single valves) combines these four principal factors within itself (that is, two steam laps and two exhaust laps), it should be noted that in the four-valve type of engine the same factors are distributed among four valves, each valve performing its own particular function in controlling the distribution of the steam for the end of the cylinder to which it is attached. Also each valve may be adjusted to a certain degree independently of the others, and this fact goes far towards explaining why engines of this type, with the disengaging valve gear, are so much more economical in the use of steam than are those with the ordinary fixed cut off. Thus, for instance, the steam valves of a Corliss engine may be adjusted to cut off the steam at any point, from the very beginning up to one-half of the stroke, without in the least affecting the release or compression, because these events are controlled by the exhaust valves.

In some of the modern improved makes of four-valve engines there are two eccentrics, one for the steam and the other for the exhaust valves. This arrangement permits of still greater latitude in adjustments for the economical use of steam.

SETTING THE VALVES OF A GREENE WHEELOCK ENGINE.

In this engine, each cylinder is equipped with four valves of the Hill gridiron type, two steam and two exhaust to each cylinder, and each valve is driven by a separate eccentric. This type of valve and gear gives a large port opening, with a minimum of travel, which in connection with the Greene cut off on the steam, and the toggle motion on the exhaust valves, gives the quickest

action at the right time to both. A minimum lap is also obtained with the aid of the gear. It is therefore very important that the movements of the valve and gear be thoroughly understood and great care be used in adjusting it. The valve plugs contain the valve seats as an integral part of the plug. These are in turn removable for repairs, as are also the valves when in position. To the plug is attached the head that holds the working parts of the valve mechanism.

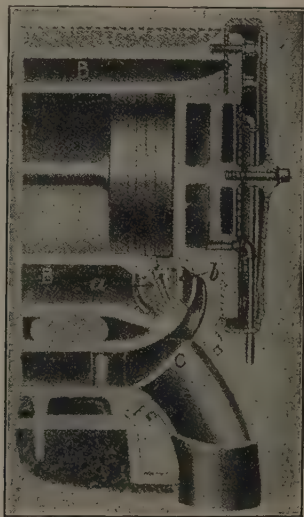


Fig. 177.

The arrangement of the valves beneath the cylinder is the Wheelock system, and allows short ports, small clearance volume and a free discharge for the water of condensation through the exhaust. The throttle also is

beneath and admits steam to the steam chest under the cylinder. There are four eccentrics, one for each valve. These eccentrics are of small size and short throw, and receive their motion from the eccentric shaft extending from the back cylinder head, alongside the engine frame to the main crank shaft, from which it receives rotary motion through bevel gear.

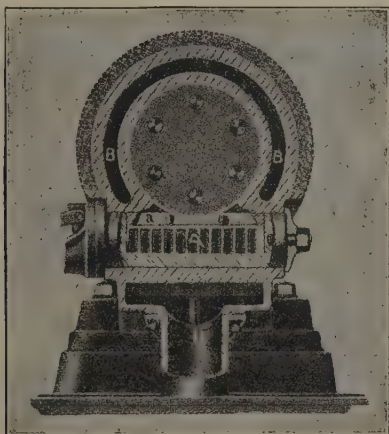


Fig. 178.

An understanding of the valve plugs and their location may be had by reference to Figs. 177, 178 and 179. Fig. 177 shows a longitudinal section of the cylinder and the cross-section of the valve plug at A. This view gives the location of the inlet (steam) valve and seat at *a* and the outlet (exhaust) valve and seat at *b*, the steam chest *B B* forming a jacket for part of the cylinder, as well as admitting the steam through the inlet *a* into the

cylinder. From the cylinder the steam passes out through the outlet *b* into the exhaust passage *C*.

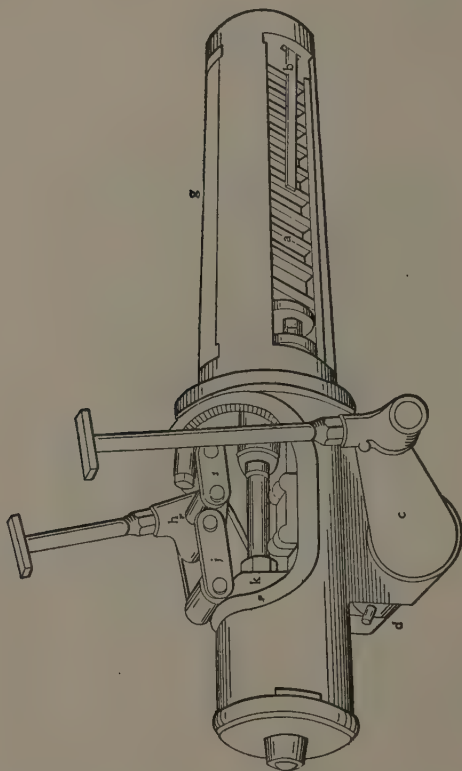


Fig. 179.

Fig. 178 is a cross-section of the cylinder through the clearance space and a longitudinal of the valve plug in

that end of the cylinder, showing the back of the inlet valve seat, with the outlet valve cut away.

Fig. 179 is a view of the valve plug with all of the parts assembled. This view shows the inlet- or steam-valve side of the plug. The inlet valve is at *a*; the spring which holds it to its seat when not under steam pressure is at *b*; and *c* is the pusher crank which actuates the valve by means of a cam at *d*, which comes in contact with the latch of the valve-stem head *e*. This is fastened to the inlet valve-stem by clamp bolts. The inlet valve-stem screws into the nut *f*, so that by loosening the clamp bolts of the head *e* and turning the rod, an adjustment of the valve setting can be made, as will be shown later.

The inlet valve is opened by the pusher cam pushing it forward, but is released from this cam through the means of a trip cam on the bottom of the valve-plug head, which is connected to the governor-rods. When released by the trip cam, the valve cuts off by means of the steam pressure on the valve-stem controlled by a dash-pot arrangement in the valve-plug head, to which the other end of the rod is attached.

The outlet valve is inside of the valve plug under the strut *g*. The position of this valve in relation to the inlet can be noted by reference to Fig. 177, where the cross-section of the valves and seats is shown. The outlet valve is actuated by the eccentric acting on the toggle joint *h*, connected between the two pairs of links, from the point *i*, where it is fixed, and the point *j*, where the link is fastened to the valve-rod head *k* on the outlet valve-stem.

INSTRUCTIONS FOR PROPER SETTING.

The following instructions are from the builders of these engines, and if adhered to will give proper setting of the valves. The previous illustrations will aid to a full understanding of these operations.

For reference and a means of checking off the action of the valves it is stated that "A-size" valves have $5/32$ -inch lap, with $3/4$ -inch travel, and are generally used on cylinders up to and including 16 inches in diameter; "B-size" valves have $3/16$ -inch lap, with $11/8$ -inch travel, and are generally used on cylinders from 18 to 26 inches in diameter, inclusive; "C-size" valves have $1/4$ -inch lap, with $13/8$ -inch travel, and are used on cylinders from 28 inches in diameter upward.

When starting to adjust the valves, first have all eccentrics loose on the cylinder shaft, and, second, determine the direction the cylinder shaft is to run, and always rotate the eccentrics in the same direction, whether loose on the shaft, or when the shaft and eccentrics turn together.

TO ADJUST THE TRAVEL OF THE STEAM VALVES.

On the edge of the pusher crank a line is made in the shop, and on the side of the plug head, next to the pusher crank, a corresponding line is made (where the arrow points). When the line on the pusher crank corresponds exactly with the line on the side of the plug head, the pusher plate is vertical. This is its most backward position.

Adjust the eccentric-rod for this valve to such a length that in turning the eccentric around on the shaft the

line on the edge of the pusher crank comes back to correspond exactly with the line on the plug head at each revolution. Then, by shimming, adjust the bridge-supporting trip cam, so that the steam valve will travel $\frac{3}{4}$ of an inch on "A-size," $1\frac{1}{8}$ inches on "B-size" and $1\frac{3}{8}$ inches on "C-size," but bear in mind that the valve must trip at the end of its travel and the bridge must not be so low that the valve will carry the full stroke without tripping. The roller of the lifter must be in position for full travel.

TO SET THE STEAM VALVES.

On the steam valve-stem four scratch lines are made. These lines represent the valve on its lap, the valve just opening, the valve wide open and the valve pushed in until it strikes the plug. With each valve-gear a steel-wire tram is sent. Just above the valve-stem on the plug-head casting a prick-punch mark will be found. Loosen up the inlet stem head on the stem, then shove the valve back until it strikes the plug. If the valve is set correctly, the tram with one end in the mark on the plug head casting should with the other end meet the first scratch line on the valve-stem (nearest the outside end of stem). If the point of the tram does not coincide with this line, the valve-stem should be screwed in or out until it does. The valve should then be let back so that the dasher strikes the head, and the inlet stem head be brought back against the pusher plate when the pusher plate is vertical, leaving $\frac{1}{64}$ inch clearance between the pusher and latch plates. It will then be found that the point of the tram will correspond with the fourth mark on the stem, with the valve closed.

When the valve is moved forward so that the tram

point corresponds with the third line on the stem, the valve is just closing or opening, and when moved farther so that it corresponds with the second line, the valve is wide open. The travel of the valve should be between the second, third and fourth points spoken of, and it should trip just as the tram point corresponds with the second line from the outside end. Then, with the piston on dead center, the eccentric should be revolved on the shaft to bring the steam valve $1/32$ of an inch open on the crank end and $3/64$ of an inch on the head end. The eccentric should then be clamped to the shaft, and the valve is set.

TO ADJUST THE EXHAUST VALVES.

On the outside of the plug head are four prick-punch marks. On the outside of the outlet stem head where the tram rests is another prick-punch mark. This is for one point of tram.

TO ADJUST VALVES FOR LAP.

The eccentric rod should be disconnected from the eccentric. Shove the valve back as far as it will go. With the valve in this position, the outside end of the tram should fall into the fourth mark on outside of the plug head nearest the cylinder. If it does not, loosen up the nut holding the outlet stem head, and screw the stem in or out sufficiently to make the tram come into the fourth mark. Then tighten up the nut holding the outlet stem head, connect the eccentric-rod to the eccentric, lengthen or shorten this eccentric rod so that the travel of the valve due to one revolution of the eccentric will

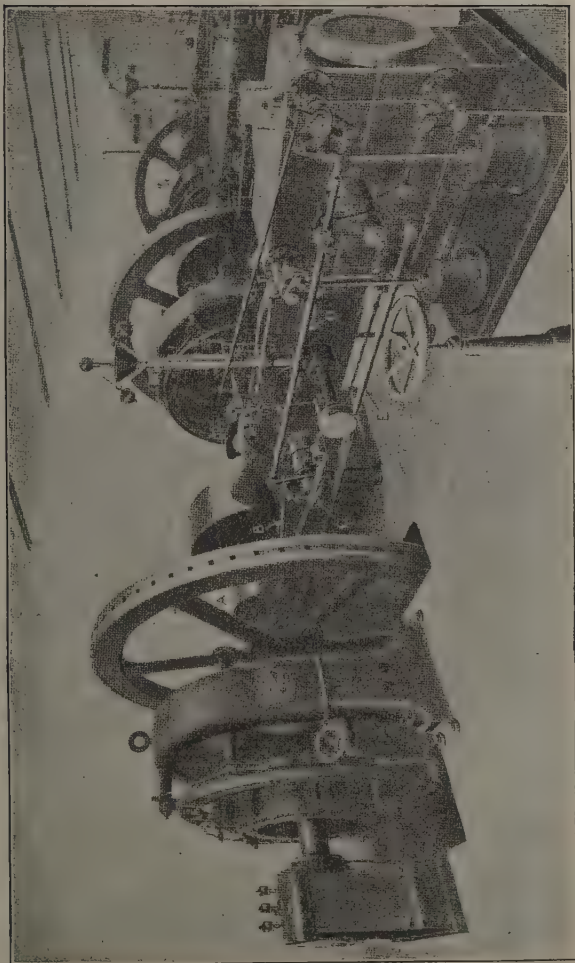


Fig. 180. Valve-Gear Side of Reynolds Long-Range Cut-Off Engine.

move the tram from the first to the third prick-punch mark; and no farther.

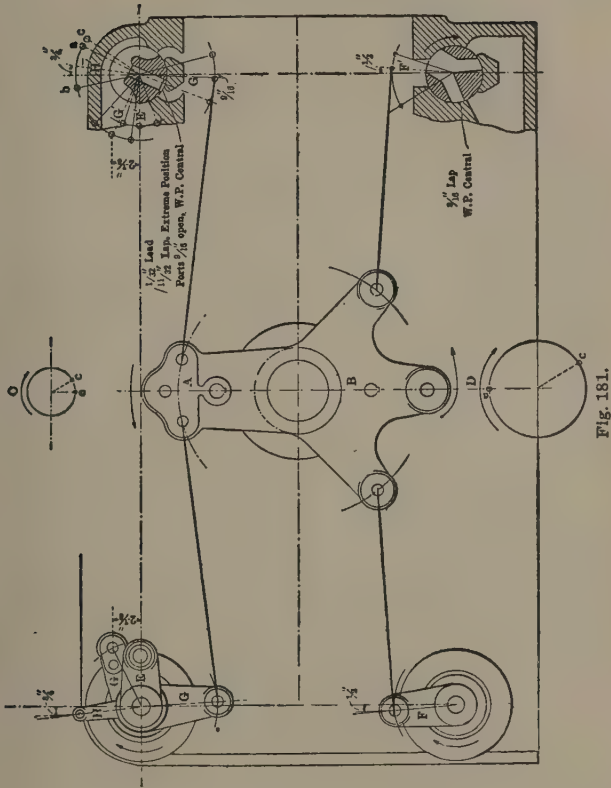
The eccentric should then be set so that when the piston is about 5 inches from the end of the return stroke, the exhaust valve should have just closed, and the tram point would fall into the second mark on the plug head.

As these valves must be set while the valves are out of sight, a strict adherence to these rules of adjustment must be followed, care being taken to be accurate on all points.

REYNOLDS LONG RANGE CUT OFF.

Fig. 180 shows the valve gear side of an Allis-Chalmers engine equipped with the long range cut off which is designed to give a maximum cut-off for power, and the essential feature of the steam valves is that they have a negative lap or opening when in mid-position, the cut-off being made entirely by the governor through the knock-off cam.

Referring to Fig. 181, the steam and exhaust valves on one end of the cylinder are shown with the valve-gear removed and the valves and ports in cross-section, while on the other end the valve-cranks have been left in place and show their relative position to the valves at the opposite end. The steam and exhaust wrist-plates are shown at *A* and *B*, respectively, and above *A* is shown the travel circle *C* of the steam eccentric; below is the exhaust circle *D*. In these circles the crank position is at *c* and *e* is the eccentric position. The steam valve-crank is indicated by *E*, the exhaust valve-crank by *F*; *G* is the bell-crank and *H* the knock-off cam. On the other end of the cylinder where the valves and ports

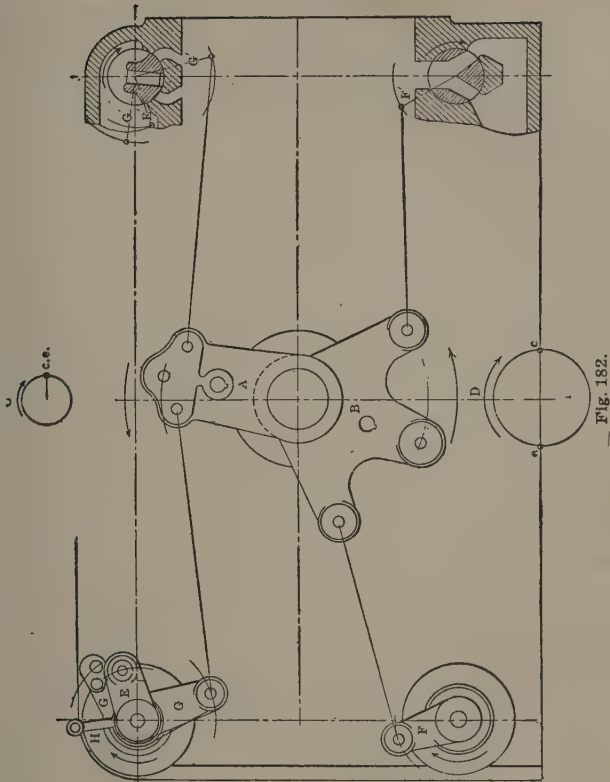


are in cross-section, the dotted lines E' , F' , G' and H' denote the center lines of the same parts on that end, and the arcs at the ends of these lines show the respective positions of the pin centers. From each end of these arcs the center lines show the positions of the pins when they reach their respective extremes of travel.

STEAM VALVES OPEN AT BOTH ENDS WHEN HOOKED UP.

In Fig. 181 the wrist-plates and all connected parts are shown in their central positions, at which the exhaust valves are lapped, as is usual in practice, but the steam valves are open on both ends when they are hooked up. If hooked up and not released the steam valves would be open from the beginning of one stroke up to 75 per cent of the return stroke, but when the knock-off cam-pin center is at *a*, the cut-off will be carried out to about seven-eighths or eleven-twelfths of the stroke, and the cut-off will occur just before the steam valve on the opposite end picks up for lead. When the knock-off cams are in the position represented by the lines *H* and *H'*, Fig. 181, the cut-off will occur at about three-eighths of the stroke, and when the knock-off pin center is at *b* the valves will remain lapped, being dropped before they can open. If the regulator is allowed to drop down so the knock-off cam-pin will reach the point *c*, the valves will not pick up and will remain lapped. This peculiarity must be thoroughly fixed in mind.

In Fig. 182 the valve-cranks are in their extreme positions, and the eccentrics likewise, with everything ready to start in the direction of the arrows. On the crank end the steam valve is lapped and the exhaust valve is open, while reverse conditions exist on the head end. On all other types of valve-gear the eccentrics would be advanced 90 degrees when the valves are lapped, but on this engine the steam valve is lapped when the eccentric is on its extreme position. The exhaust valves are the same as on any other double-eccentric Corliss engine.



SETTING THE VALVES.

Bearing these points in mind we may proceed to set the valves. The amounts of lap and lead and the positions of the cranks from the center lines given herewith are for engine cylinders of 36, 42, 48, and 60-inch stroke.

First set the wrist-plates central and clamp them in place; then adjust the lengths of the rods so that the steam valves are open $11/32$ inch, as shown in Fig. 181, and the exhaust valves are lapped $3/16$ inch. If the rod lengths are right the center lines of the cranks E and E' will coincide, the pins of the cranks F and F' will be one-half inch from the center line, as shown, and the pins on each end of the bell-cranks G and G' will be $27/8$ inches and $9/16$ inch from the center lines. When the valves have been set with the wrist-plates central, release the wrist-plates and roll the eccentrics around the shaft to test them and the reach-rods, and see that they are of the right length to make the wrist-plate travel equally each side of the center line.

Then place the crank on center and pull the steam eccentric around enough to give $1/32$ -inch lead, and make it fast. Next move the engine around in its direction of travel to about 95 degrees of its stroke and move the exhaust eccentric around until the exhaust valve on the same end is just opening or releasing. Make the exhaust eccentric fast and move the engine around its full revolution and check off the valves on the other end and the exhaust closure. Then set the regulator up to its central position and adjust the lengths of the rods from the lever to the knock-off cams, so that the pins of the cams H and H' will set $3/4$ inch off the center line, as in Fig. 181. Let the regulator down and hook up the wrist-plates; then pull the engine around to make sure that the steam valves are released on each stroke alternately at not later than eleven-twelfths of the stroke, and always before the other valve picks up.

SHAFT GOVERNORS.

Many automatic cut-off engines, especially those of the high speed type, are fitted with isochronal or shaft governors. There are various styles of these governors, but all or nearly all of them control the admission of steam to the cylinder, and consequently the point of cut off by varying the angular advance of the eccentric, which in such engines is free to move across the shaft, being entirely under the control of the governor.

Very close regulation is generally obtained by the use of shaft governors, but particular attention should be given to the lubrication of the steam valve, which, with this class of engines, is generally a slide valve of some description, and although it may be ever so nicely balanced, yet if it does not get sufficient oil, the friction, due to dry surfaces rubbing together, will put extra work on the governor, and the speed is liable to be irregular.

Keying Up. To keep the working parts of an engine properly keyed up so as to take up the lost motion without causing the bearings to heat, is one of the most delicate and exacting duties of the engineer. Every engineer worthy of the name should aim to have a smooth and quietly running engine. In fact, this is one of the principal tests of his skill as an engineer.

A few general suggestions may be given here, but much more depends upon the good judgment of the engineer himself.

In connecting up an engine, as for instance, the crank pin and crosshead, the key should first be driven in until it is solid, that is, until the brasses clamp the pin tightly. Then place a small ruler across the face

of the gib and key, and with the point of a knife blade, or a steel scriber, make a fine mark along the edge of the ruler. This will be the "solid mark." Now back the key up by light blows with the hammer, which should be of copper, until the brasses are just loose enough on the pin to permit a side movement of the rod. If the rod is very heavy, hold a block of wood against the side of the rod near the pin and give it a

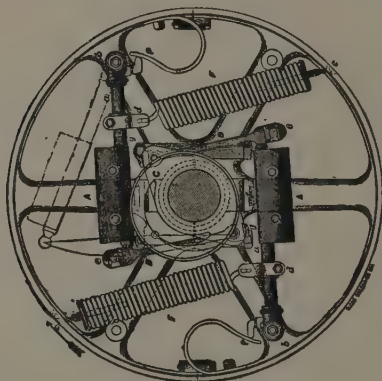


Fig. 183. Isochronal or Shaft Governor.

blow with a sledge. If there is no movement, back the key a little more, and keep trying until there is a side movement. There should be a space of at least $\frac{1}{32}$ in. between the sides of the brasses and the flange of the pin, so as to allow a slight side play. If it is found, after running a while, that there is too much lost motion, causing the engine to pound, put the ruler across the gib and key again, and with a lead pencil draw a line. Then loosen the set screw and drive the

key in a distance equal to the width of the line, and no more. This process should be repeated at intervals until the pound is all gone, or the bearing begins to warm up slightly, indicating that the brasses are up as close as they will run.

The adjustment of pillow blocks also requires a large measure of skill and good judgment, and should be done by degrees; but when once adjusted properly and kept well oiled, the matter of keeping them in good working condition becomes greatly simplified.

If a pillow block, or other bearing filled with babbit metal, should become heated to such a temperature as to cause the metal to run, do not shut down the engine at once and turn on a stream of cold water, because this will only make matters worse, causing the metal to stick to the journal, and the labor and trouble of cleaning it out will be increased. The best method to pursue in such an emergency is to gradually slow the engine down and keep it moving at as slow a speed as it will run, and in the meantime a small stream of cold water may be allowed to flow over the bearing, applying it to all parts as nearly as possible until it is cooled, after which the engine may be stopped and repairs made.

THE STEAM TURBINE.

Although the turbine principle of utilizing the heat energy in steam and converting it into useful work has been experimented upon for many years, it is only since the inauguration of the twentieth century that steam turbines have been brought to the front as efficient power producers.

The piston of the reciprocating engine is driven back and forth by the static expansive force of the steam, while in the steam turbine not only the expansive force is made to do work, but a still more important element is utilized, viz., the kinetic energy, or heat energy latent in the steam, and which manifests itself in the rapid vibratory motion of the particles of steam expanding from a high to a low pressure, and this motion the steam turbine transforms into work.

WESTINGHOUSE-PARSONS STEAM TURBINE.

The Westinghouse-Parsons Steam Turbine is fundamentally based upon the invention of Mr. Charles A. Parsons, who, while experimenting with a reaction turbine constructed along the lines of Hero's engine, conceived the idea of combining the two principles, reaction and impulse, and also of causing the steam to flow in a general direction parallel with the shaft of the turbine. This principle of parallel flow is common to all four types of turbines, but is perhaps more prominent in the Westinghouse-Parsons and less so in the De Laval.

A cubic foot of steam under 100 lbs. pressure, if allowed to discharge into a vacuum of 28 in., would attain a theoretical velocity of 3,860 ft. per second and would exert 59,900 ft.-lbs. of energy. A law of turbo mechanics specifies that in order to obtain the highest efficiency in the operation of turbines (whether water or steam) the relation between bucket speed and fluid speed (steam in this case) should be as follows:

For purely impulse wheels, bucket speed equals one-half of jet speed.

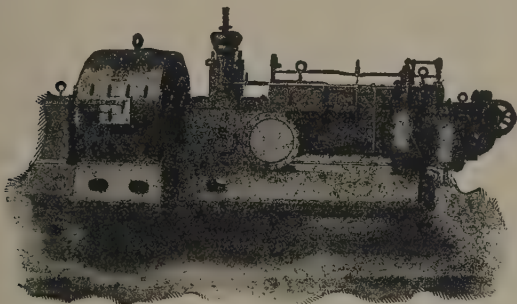


Fig. 184.

For reaction wheels, bucket speed equals jet speed.

Assuming the velocity of the jet of steam issuing from the nozzle to be 4,000 ft. per second, this would mean a peripheral speed of 2,000 ft. per second for an impulse wheel, and for a wheel 1 ft. in diameter the speed would be 38,100 R. P. M. But such a speed is beyond the limits of strength of material.

As before stated, the Westinghouse-Parsons turbine operates on both impulse and reaction principles, and by a system of compounding, which will be explained

later on, the peripheral velocity of the machine has been so reduced as to bring it within practical limits while at the same time the power value of the steam is utilized to a high degree of efficiency.

The speed of the Westinghouse-Parsons turbine varies from about 750 R. P. M. for a 5,000 K. W. machine to 3,600 R. P. M. for a 400 K. W. turbine.

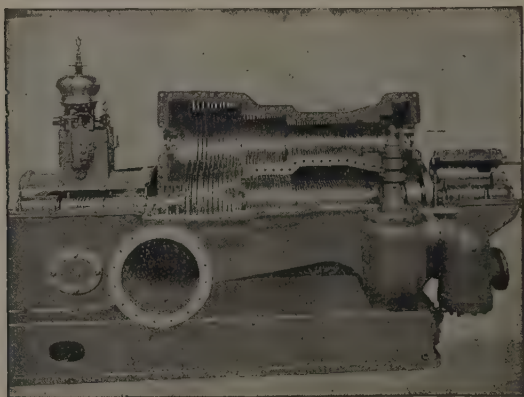


Fig. 185.

Fig. 184 is a general view of a 400 K. W. turbine generator unit. Fig. 185 shows a 600 H. P. machine with the upper half of the cylinder, or stator as it is termed, thrown back for inspection. Fig. 186 is a sectional view of a Westinghouse-Parsons turbine, and it will be noticed that there are three sections or drums, gradually increasing in diameter from the inlet A to the third and last group of blades. This arrangement may be likened in some measure to the triple compound reciprocating engine.

By reference to Fig. 185 it will be seen that the inside of the cylinder is studded with rows of small stationary blades and that the rotor or revolving part of the machine is also fitted with rows of small blades, similar in shape and dimensions to the stationary blades. When the upper half of the cylinder is in position, each row of stationary blades fits in between two corresponding rows of moving blades. This arrangement may perhaps be better understood by reference to Fig. 187, which illustrates the relation of the stationary blades to the moving blades when in position, and also shows by the arrows the course of the steam and its change of direction caused by the stationary blades.

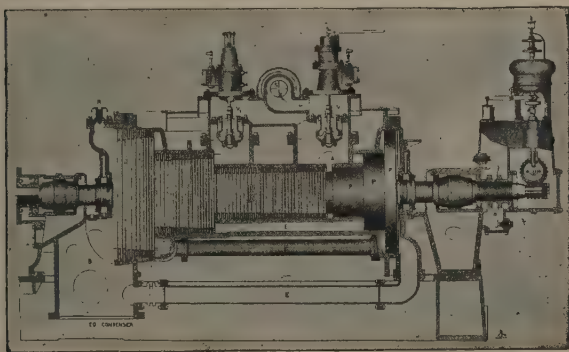


Fig. 186.

For the purpose of explanation the moving blades or vanes may be considered as small curved paddles projecting from the surface of the rotor, and there is a large number of them, as, for instance, taking a 400 K. W. machine, there are 16,095 moving blades and 14,978 stationary blades, a total of 31,073.

The stationary vanes, as previously explained project from the inside surface of the cylinder. Both stationary and moving vanes are similar in shape, and are made of hard drawn material, and they are set into their places and secured by a caulking process. The blades vary in size from $\frac{1}{2}$ to 7 in. in length, according to where they are used. Referring to Fig. 185, it will be observed that the shortest blades are placed at what might be termed the steam end of each section or drum of the rotor and cylinder, and that their length gradually increases, corresponding with the increased volume of steam, until a mechanical limit is reached, when a

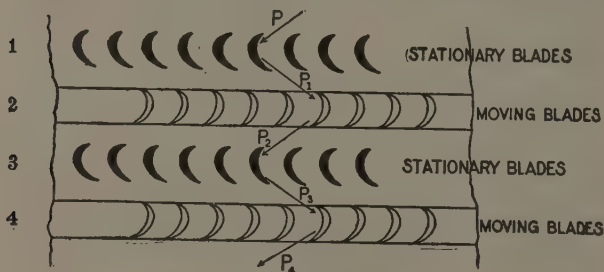


Fig. 187.

new group of blades begins on a succeeding drum of larger diameter. Referring to Fig. 187, which is a sectional view of four rows of blades, it will be noticed that all the blades, whether stationary or moving, have the same curvature. Also that the curves are set opposite each other. The reason for this will be apparent as the diagram is studied. The steam at pressure P first comes in contact with row 1 of stationary blades. It expands through this row, and in expanding the pressure falls to P' .

The energy in the steam is converted into velocity, and it impinges upon row 2 of moving blades, driving them around in their course by impulse. A second expansion now occurs in row 2 and again the energy is converted into velocity, but this time the reaction of the steam as it leaves the blades of row 2 also tends to impel them around in their course. The moving blades thus receive motion from two causes—the one due to the impulse of the steam striking them, and the other due to the reaction of the steam leaving them.

This cycle is repeated in rows 3 and 4, and so on throughout the length of the rotor until the exhaust end is reached.

It should be noted that the general direction taken by the steam in its passage through the turbine is in the form of a spiral or screw line about the rotor. The clearance between the blades as they stand in the rows is $\frac{1}{8}$ in. for the smallest size blades and $\frac{1}{2}$ in. for the larger ones, gradually increasing from the inlet to the exhaust. In the 5,000 K. W. machine the clearance at the exhaust end between the rows of blades is 1 in. It will thus be seen that there is ample mechanical clearance, also allowance for lateral motion for adjustment of the rotor, although this is very slight, as the rotor is balanced at all loads and pressures by the balancing pistons PPP, Fig. 186, to which reference is now made. These pistons revolve within the cylinder, but do not come in mechanical contact with it; consequently there is no friction. The diameter of each piston corresponds to the diameter of one of the three drums.

The steam entering the chamber A through valve V presses against the turbine blades and goes through doing work by reason of its velocity. It also presses

equally in the opposite direction against the first piston P, and so the shaft or rotor has no end thrust. On leaving the first group of blades and striking the second group the pressure in either direction is again equalized by the balance port E allowing the steam to press against the second balance piston P. The same event occurs at group three, the steam acting upon the third piston P.

The areas of the balancing pistons are such that, no matter what the load may be or what the steam pressure or exhaust pressure may be, the correct balance is maintained and there is practically no end thrust. Below is shown a pipe E connecting the back of the balancing pistons with the exhaust chamber. This arrangement is for the purpose of equalizing the pressure at this point with the pressure in the exhaust chamber B.

It might be thought that the blades, on account of their being so light and thin, would wear out very fast, but experience so far shows that they do not. This may be accounted for in two ways. First, the reduction of the velocity of the steam, the highest velocity in the Parsons turbine not exceeding 600 ft. a second; secondly, the light steam thrust on each blade, said to be equal to about 1 oz. avoirdupois. This is far within the bending strength of the material. A steam strainer is also placed in the admission port, to prevent all foreign substances from entering the turbine.

A rigid shaft and thrust or adjustment bearing accurately preserves the clearances, which are larger in this turbine than in other types, owing to the fact that the entire circumference of the turbine is constantly filled with working steam when in operation.

The bearings shown in Fig. 186 are constructed along lines differing from those of the ordinary reciprocating

engine. The bearing proper is a gun metal sleeve that is prevented from turning by a loose-fitting dowel. Outside of this sleeve are three concentric tubes having a small clearance between them. This clearance is kept constantly filled with oil supplied under light pressure, which permits a vibration of the inner shell or sleeve and at the same time tends to restrain or cushion it. This arrangement allows the shaft to revolve about its axis of gravity, instead of the geometrical axis, as would be the case if the bearing were of the ordinary construction. The journal is thus to a certain degree a floating journal, free to run slightly eccentric according as the shaft may happen to be out of balance.

A flexible coupling is provided, by means of which the power of the turbine is transmitted to the dynamo, or other machine it is intended to run. The oil from all the bearings drains back into a reservoir, and from there it is forced up into a chamber, where it forms a static head, which gives a constant pressure of oil on all the bearings. A secondary valve is located at Vs, by means of which high pressure steam may be admitted to the steam space E on the same principle that high pressure steam is admitted to the low pressure cylinder of a compound engine. This valve opens automatically in cases of emergency, such as overload, failure of the condenser to work, etc.

The shaft, where it passes through either cylinder head, is packed with a water seal packing, consisting of a small paddle wheel attached to the shaft, which, through centrifugal action, maintains a static pressure of about 5 lbs. per sq. in. in the water seal, thus preventing all leakage while at the same time it is frictionless.

The speed of the Westinghouse-Parsons turbine is regulated by a fly ball governor constructed in such manner that a very slight movement of the balls serves to produce the required change in the supply of steam. Fig. 188 is a diagram of the governor mechanism. The ball levers swing on knife edges instead of pins. The governor works both ways, that is to say, when the levers are oscillating about their mid position a head of steam corresponding to full load is being admitted to the turbine, and a movement from this point, either up or down, tends to increase or to decrease the supply of steam.

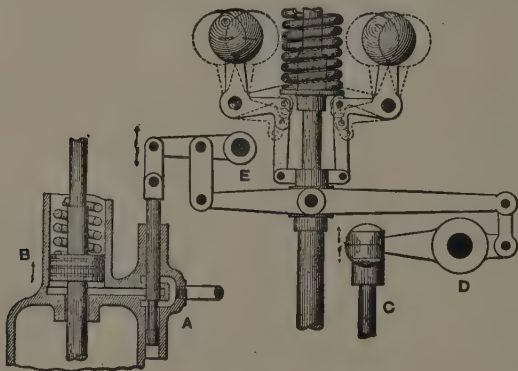


Fig. 188.

Referring to Fig. 188, B is a piston directly connected to the admission valve. Steam is admitted to this piston under control of the pilot valve A, which has a slight but continuous reciprocating motion derived from the eccentric rod C, and the function of the governor is to vary the plane of oscillation of this valve, thus

causing it to admit more or less steam to piston B. The admission valve, being actuated exclusively by piston B, is thus caused to remain open for a longer or shorter period of time, according to the load upon the turbine.

The vibrations of the admission valve, although very slight, are continuous and regular, about 165 per minute, and are transmitted primarily by means of an eccentric, the rod of which is shown at C, Fig. 188.

The governor sleeve is used as a floating fulcrum, and the points D and E are fixed. By means of this very ingenious device the steam is admitted to the turbine in puffs, either long or short, according to the demand for steam. At full load the puffs merge into an almost continuous blast. When the load has increased to the point where the valve is wide open continuously, a full head of steam is being admitted. Beyond this the secondary valve comes into action, thus keeping the speed up to normal.

The rotor, requires perfect balancing to insure quiet running, but this is easily accomplished in the shop by means of a balancing machine used by the builders.

Steam turbines generally show higher efficiency in the use of steam than reciprocating engines do, and this fact is due to three leading causes. First, it is possible with the turbine to use highly superheated steam which, owing to the difficulties attending lubrication, could not be used in the reciprocating engine. Second, a larger proportion of the heat contained in the steam is converted into work, for the reason that the steam is allowed to expand to a much lower pressure and into a higher vacuum. In addition to this, the velocity of the expanding steam is utilized in a much higher degree in the turbine as compared with the re-

ciprocating engine. Third, mechanical friction or lost work is reduced to the minimum. Under test a 400 K. W. Westinghouse-Parsons steam turbine, using steam at 150 lbs. initial pressure and superheated about 180° , consumed 11.17 lbs. of steam per brake horse power hour at full load. The speed was 3,550 R. P. M. and the vacuum was 28 in. With dry saturated steam the consumption was 13.5 lbs. per B. H. P. hour at full load, and 15.5 lbs. at one-half load.

A 1,000 K. W. machine, using steam of 150 lbs. pressure and superheated 140° , exhausting into a vacuum of 28 in., showed the very remarkable economy of 12.66 lbs. of steam per E. H. P. per hour.

A 1,500 K. W. Westinghouse-Parsons turbine, using dry saturated steam of 150 lbs. pressure with 27 in. vacuum, consumed 14.8 lbs. steam per E. H. P. hour at full load, and 17.2 lbs. at one-half load.

CURTIS STEAM TURBINE.

In the Curtis turbine the heat energy in the steam is imparted to the wheel, both by impulse and reaction, but the method of admission differs from that of the Westinghouse-Parsons in that the steam is admitted through expanding nozzles in which nearly all of the expansive force of the steam is transformed into the force of velocity. The steam is caused to pass through one, two, or more stages of moving elements, each stage having its own set of expanding nozzles, each succeeding set of nozzles being greater in number and of larger area than the preceding set. The ratio of expansion within these nozzles depends upon the number of stages, as, for instance, in a two-stage machine, the steam enters the initial set of nozzles at boiler pressure, say 180 lbs.

It leaves these nozzles and enters the first set of moving blades at a pressure of about 15 lbs., from which it further expands to atmospheric pressure in passing through the wheels and intermediates. From the pressure in the first stage the steam again expands through



Fig. 189. 5,000 K. W. Curtis Steam Turbine Direct Connected to 5,000 K. W. Three-Phase Alternating Current Generator.

the larger area of the second stage nozzle to a pressure slightly greater than the condenser vacuum at the entrance to the second set of moving blades, against which it now impinges and passes through still doing work, due to velocity and mass.

From this stage the steam passes to the condenser. If the turbine is a four-stage machine and the initial pressure is 180 lbs., the pressure at the different stages would be distributed in about the following manner: Initial pressure, 180 lbs.; first stage, 50 lbs.; second stage, 5 lbs.; third stage, partial vacuum, and fourth stage, condenser vacuum.

Fig. 189 gives a general view of a 5,000 K. W. turbine and generator. The generator is shown at the top, while the turbine occupies the middle and lower section. A portion of the inlet steam pipe is shown, ending in one nozzle group at the side. There are three groups of initial nozzles, two of which are not shown. The revolving parts of this unit are set upon a vertical shaft, the diameter of the shaft corresponding to the size of the unit. For a machine having the capacity of the one illustrated by Fig. 189 the diameter of the shaft is 14 in.

The shaft is supported by and runs upon a step bearing at the bottom. This step bearing consists of two cylindrical cast iron plates, bearing upon each other and having a central recess between them into which lubricating oil is forced under pressure by a steam or electrically driven pump, the oil passing up from beneath. A weighted accumulator is sometimes installed in connection with the oil pipe as a convenient device for governing the step bearing pumps, and also as a safety device in case the pumps should fail, but it is seldom required for the latter purpose, as the step bearing pumps have proven, after a long service in a number of cases, to be reliable. The vertical shaft is also held in place and kept steady by three sleeve bearings, one just above the step, one between the

turbine and generator, and the other near the top. These guide bearings are lubricated by a standard gravity feed system. It is apparent that the amount of friction in the machine is very small, and as there is no end thrust caused by the action of the steam, the relation between the revolving and stationary blades may be maintained accurately. As a consequence, therefore, the clearances are reduced to the minimum.

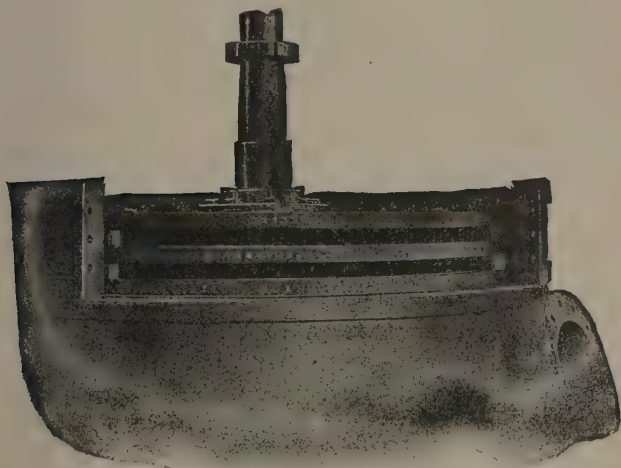
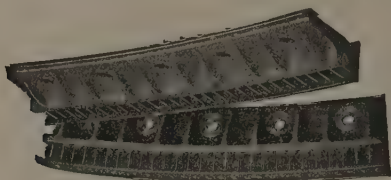


Fig. 190. 500 K. W. Curtis Steam Turbine in Course of Construction.

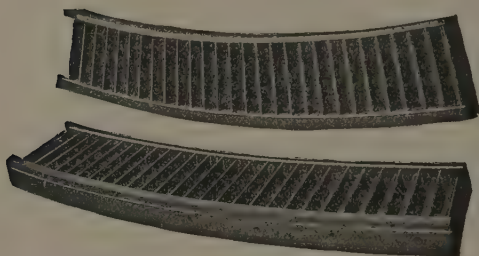
The Curtis turbine is divided into two or more stages, and each stage has one, two or more sets of revolving blades bolted upon the peripheries of wheels keyed to the shaft. There are also the corresponding sets of stationary blades, bolted to the inner walls of the cylinder or casing. As in the Westinghouse-Parsons type,

the function of the stationary blades is to give direction to the flow of steam.

Fig. 190 illustrates one stage of a 500 K. W. turbine in course of construction. It will be observed that there are three wheels, and that in the spaces between



Revolving Buckets for Curtis Steam Turbine.



Stationary Buckets for Curtis Steam Turbine.

Fig. 191.

these wheels the stationary buckets or vanes are placed, being firmly bolted to the casing. Fig. 191 shows sections of both revolving and stationary buckets ready to be placed in position. The illustration in Fig. 190 shows the lower or last stage. The clearance between the revolving and stationary blades is from $1/32$ to $1/16$ in., thus reducing the wastage of steam to a very low percentage. The diameters of the wheels vary accord-

ing to the size of the turbine, that of a 5,000 K. W. machine being 13 ft.

Fig. 192 shows a nozzle diaphragm with its various openings, and it will be noted that the nozzles are set at an angle to the plane of revolution of the wheel.

Fig. 193 is a diagram of the nozzles, moving blades and stationary blades of a two-stage Curtis steam turbine. The steam enters the nozzle openings at the top, controlled by the valves shown, the regulation of which will be explained later on. In the cut Fig. 193 two of the valves are open, and the course of the steam through the first stage is indicated by the arrows. After passing successively through the different sets of moving blades and stationary blades in the first stage, the steam

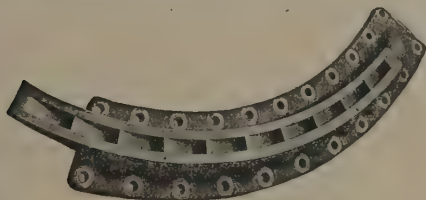


Fig. 192. Nozzle.

passes into the second steam chest. The flow of steam from this chamber to the second stage of buckets is also controlled by valves, but the function of these valves is not in the line of speed regulation but for the purpose of limiting the pressure in the stage chambers, in a manner somewhat similar to the control of the receiver pressure in a two-cylinder or three-cylinder compound reciprocating engine.

The valves controlling the admission of steam to the second and later stages differ from those in the first

group in that they partake more of the nature of slide valves and may be operated either by hand or automatically; in fact, they require but very little regulation,

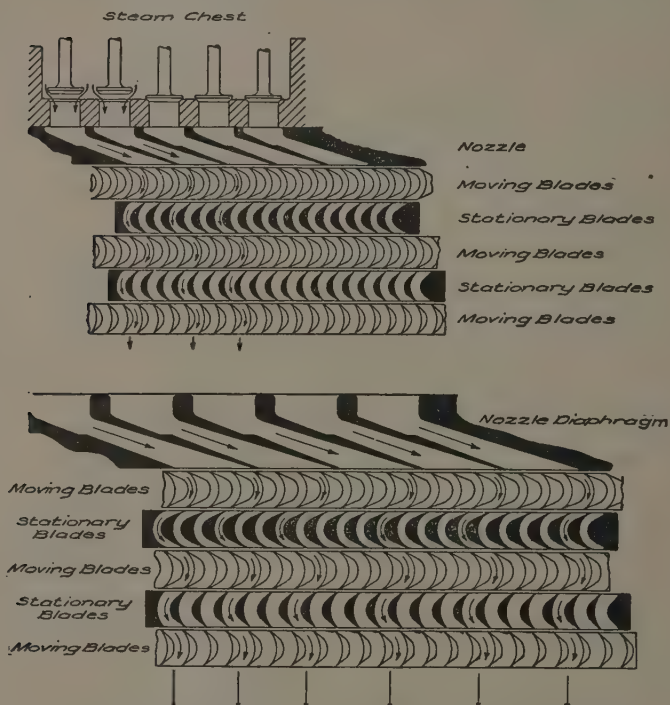


Fig. 193. Diagram of Nozzles and Buckets in Curtis Steam Turbine.

as the governing is always done by the live steam admission valves.

Action of the Steam in a Two-stage Machine.—As

previously stated, the steam first strikes the moving blades in the first stage of a two-stage machine at a pressure of about 15 lbs. above atmospheric pressure, but with great velocity. From this wheel it passes to the set of stationary blades between it and the next lower wheel. These stationary blades change the direction of flow of the steam and cause it to impinge the buckets of the second wheel at the proper angle.

This cycle is repeated until the steam passes from the first stage into the receiving chamber or steam chest for the second stage. Its passage from this chamber into the second stage is controlled by valves, which, as before stated, are regulated either by hand or automatically. The course of the steam through the nozzles and blades of the second stage is clearly indicated by the arrows, and it will be noted that steam is passing through all the nozzles.

At this point it might be well to consider the question which no doubt arises in the mind of the student in his efforts to grasp the underlying principles in the action of the steam turbine. Why is it that the impingement of the steam, at so low a pressure, against the blades or buckets of the turbine, imparts such a large amount of energy to the shaft?

The answer is, because of velocity, and a good example of the manner in which velocity may be made to increase the capacity of an agent to do work is illustrated in the following way: Suppose that a man is standing within arm's length of a heavy plate glass window and that he holds in his hand an iron ball weighing 10 lbs. Suppose the man should place the ball against the glass and press the same there with all the energy he is capable of exerting. He would make very little, if

any, impression upon the glass. But suppose that he should walk away from the window a distance of 20 ft. and then exert the same amount of energy in throwing

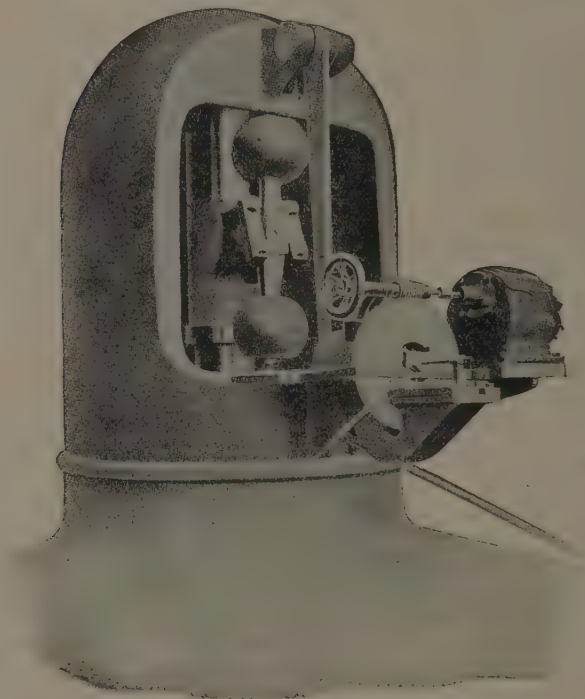


Fig. 194. Governor for 5,000 K. W. Turbine.

the ball against the glass, a different result would ensue. The velocity with which the ball would impinge the surface of the glass would no doubt ruin the window.

Now, notwithstanding the fact that weight, energy and time involved were exactly the same in both instances, yet a much larger amount of work was performed in the latter case, owing to the added force imparted to the ball by the velocity with which it impinged against the glass.

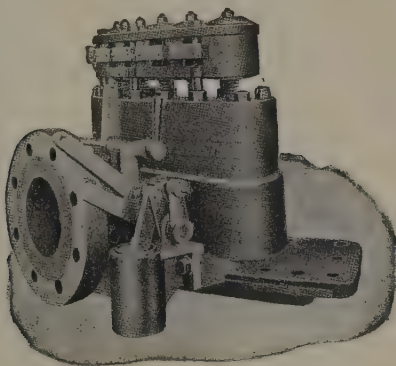


Fig. 195. Electrically Operated Valve.

Speed Regulation.—The governing of speed is accomplished in the first set of nozzles, and the control of the admission valves here is effected by means of a centrifugal governor attached to the top end of the shaft. This governor, by a very slight movement, imparts motion to levers, which in turn work the valve mechanism. The admission of steam to the nozzles is controlled by piston valves, which are actuated by steam from small pilot valves which are in turn under the control of the governor. Fig. 194 shows the form of governor for a 5,000 K. W. turbine, and Fig. 195 shows the electrically operated admission valves for one set of nozzles.

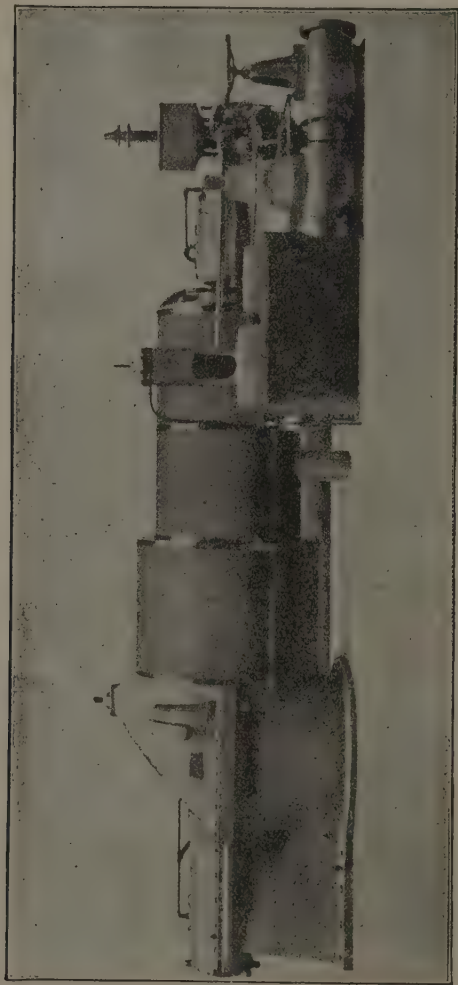


Fig. 196. The Allis Chalmers Steam Turbine.

Speed regulation is affected by varying the number of nozzles in flow, that is for light loads fewer nozzles are open and a smaller volume of steam is admitted to the turbine wheel, but the steam that is admitted impinges the moving blades with the same velocity always, no matter whether the volume be large or small. With a full load and all the nozzle sections in flow, the steam passes to the wheel in a broad belt and steady flow.

ALLIS-CHALMERS STEAM TURBINE.

Fig. 196 shows a general view of the Allis-Chalmers steam turbine, and although it is essentially of the "Parsons" type, still there are a number of modifications in details of construction, as compared with the Westinghouse-Parsons steam turbine, some of which, no doubt may be considered as adding to the efficiency, and durability of the machine.

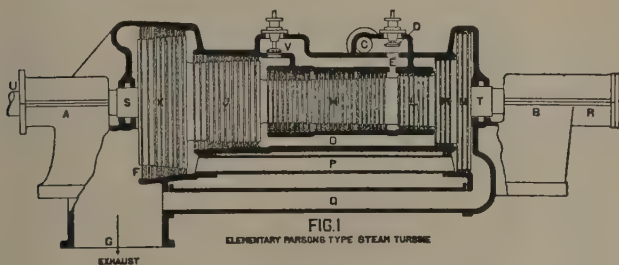


Fig. 197.

Fig. 197 is a sectional view of the "elementary" Parsons type of steam turbine, and its various parts are described as follows:

Main bearings, A and B. Thrust bearing, R. Steam pipe C. Main throttle valve, D, which is balanced, and

operated by the governor. Steam enters the cylinder through passage E, passes to the left through the alternate rows of stationary and revolving blades, leaving the cylinder at F and passes into the condenser, or atmosphere through passage G-H, J and K are the three steps or stages of the machine. L, M and N are the three balance pistons. O, P and Q are the equalizing passages, connecting the balance pistons with the corresponding stages.

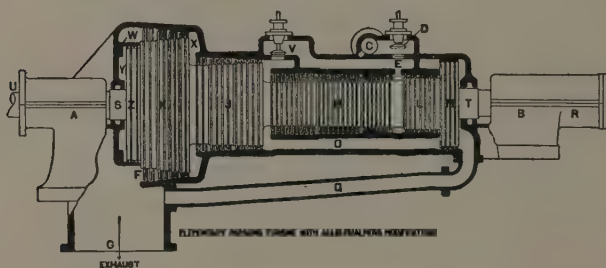


Fig. 198.

Fig. 198 shows a sectional view of the "Parsons" turbine with the Allis-Chalmers modifications. L and M are the two balance pistons at the high pressure end. Z is a smaller balance piston placed in the low pressure end, yet having the same effective area as did the larger piston N shown in Fig. 197. O and Q are the two equalizing passages for pistons L and M. Passage P is omitted in this construction and balance piston Z is equalized with the third stage pressure at Y. Valve V is a by-pass valve to allow of live steam being admitted to the second stage of the cylinder in case of a sudden overload. This by-pass valve is the equivalent of the by-pass valve used to admit live steam to the low pressure cyl-

inder of a compound reciprocating engine. Valve V is arranged to be operated, either by the governor or by hand, as the conditions may require. Frictionless glands made tight by water packing are provided at S and T where the shaft passes out of the cylinder. The shaft is extended at U and connected to the generator shaft by a flexible coupling.

The action of the steam, and the general arrangement of the stationary and moving blades is practically the same in the two turbines, with the exception that, in the larger sizes of the Allis-Chalmers turbine the "balance" pistons for neutralizing the end thrust, are arranged in a different manner, the largest one of the three pistons (piston N—Fig. 197), is replaced by a smaller balance piston.

This piston presents the same effective area for the steam to act upon, as did the larger piston, for the reason that the working area of the latter in its original location consisted only of the annular area, included between its periphery, and the periphery of the next smaller piston. The pressure of the steam is brought to bear upon this equalizing piston in its new position, by means of passages or ports through the body of the rotor, connecting the third stage of the cylinder with the supplementary cylinder, in which the piston revolves. Fig. 199 shows the arrangement of blading, the course of the steam being indicated by the arrows. The clearances between the edges of the revolving and stationary blades, as shown in the cut are relatively out of proportion to the actual clearances allowed.

This clearance is preserved by means of a small thrust-bearing provided inside the housing of the main bearing.

This thrust-bearing can be adjusted to locate and hold

the rotor in such a position as will allow sufficient clearance to prevent actual contact between the moving and stationary blades and yet reduce the leakage of steam to a minimum.

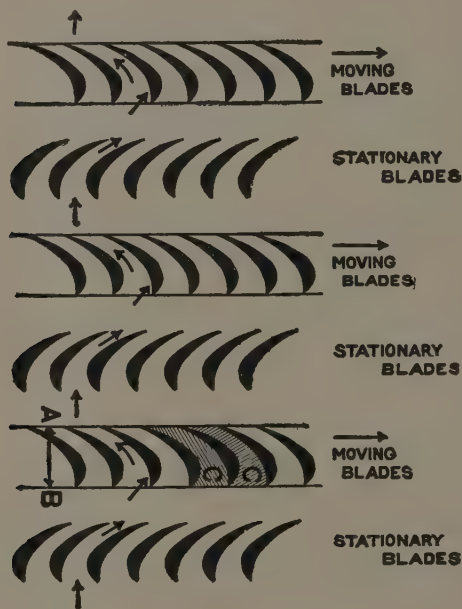


Fig. 199 showing arrangement of blading and course of the steam in Parsons steam turbine.

The method by which the blades are fitted to and held in the rotor and cylinder of the Allis-Chalmers steam turbine is as follows: Each blade is individually formed by special machine tools, so that its root or foot is of an angular, or dove-tail shape, and at its tip there is a

projection. In order that the roots of the blade may be firmly held in position, a foundation ring, A, Fig. 200, is provided, which after being formed to a circle of the proper diameter, has slots cut in it by a special milling machine.

These slots are formed of dove-tail shape to receive the roots of the blades and are at the same time accurately spaced, and inclined so as to give the required pitch and angles to the blades.



Fig. 200.

The foundation rings are also of dove-tail shape in cross-section, those holding the stationary blades are inserted in dove-tail grooves in the cylinder and those holding the revolving blades being pressed into the rotor or spindle.

The rings are firmly held in their places by key-pieces driven into place and upset into under-cut grooves, thus positively locking the whole structure together, and making it practically impossible for a blade to get out of place.

The tips of the blades are held and firmly bound together by a shroud-ring, B, Fig. 200.

The shroud-rings are made channel-shape, in cross-section, the flanges being made thin in order to prevent dangerous heating in case of accidental contact with either the walls of the cylinder or the surface of the rotor.

The bearings of this turbine are of the self-adjusting ball and socket type, designed for high speed. Shims are provided for proper alignment. The lubrication of the four bearings, two for the turbine, and two for the generator, is accomplished by supplying an abundance of oil to the middle of each bearing and allowing it to flow out at the ends where it is caught, passed through a cooler, and pumped back to the bearings.

The fact that the oil is supplied in large quantities to the bearings does not involve a heavy oil bill.

The journals are practically floating on films of oil, thus preventing that "wearing out" of the oil that occurs when it is supplied in small "doses."

The governor is driven from the turbine shaft by means of cut gears working in an oil-bath.

The governor operates a balance throttle valve by means of a relay, except in very small sizes in which the valve is worked direct.

In order to provide for any possible accidental derangement of the main governing mechanism, an entirely separate safety or over-speed governor is fur-

nished. This governor is driven directly by the turbine shaft without the intervention of gearing, and is so arranged and adjusted that if the turbine should reach a predetermined speed above that for which the main governor is set, the safety governor will come into action and trip a valve, shutting off the steam and stopping the turbine. A strainer is provided through which the steam is passed before admission to the turbine.

For connecting the rotors of the turbine and generator a special type of flexible coupling is used to provide for any slight inequality in the wear of the bearings, to permit axial adjustment of the turbine spindle, and to allow for differences in expansion. This coupling is so made that it can be readily disconnected for the removal of the turbine spindle or of the revolving field of the generator. Provision is made for ample lubrication of the adjoining faces of the coupling. The coupling is enclosed in the bearing housing, so that it is completely protected against damage, and cannot cause injury to the attendants.

Waste of heat by radiation is prevented in the following manner:

The hot parts of the turbine, up to the exhaust chamber, are covered with an ample thickness of non-conducting material and lagged with planished steel.

For large Allis-Chalmers turbines the bedplate is divided into two parts, one carrying the low-pressure end of the turbine and the bearings of the generator, the other carrying the high-pressure end of the turbine. The turbine is secured to the former, while the latter is provided with guides which permit the machine to slide back and forth with differences of expansion caused by varying temperature, at the same time maintaining the alignment.

It may be said in general of the steam turbine, that it has passed the experimental stage, and has come to the front as an efficient power producer, having a bright future before it. It has solved the problem of using superheated steam, owing to the absence of all rubbing parts exposed to the steam. This permits the use of steam of high temperature thus making it possible to realize the advantages of economical operation.

DISPOSAL OF THE EXHAUST STEAM OF STEAM TURBINES.

As in the case of the reciprocating engine, the highest efficiency in the operation of the steam turbine is obtained by allowing the exhaust steam to pass into a condenser, and experience has demonstrated that it is possible to maintain a higher vacuum in the condenser of a turbine than in that of a reciprocating engine. This is due, no doubt, to the fact that in the turbine the steam is expanded down to a much lower pressure than is possible with the reciprocating engine.

The condensing apparatus used in connection with steam turbines may consist of any one of the modern improved systems, and as no cylinder oil is used within the cylinder of the turbine, the water of condensation may be returned to the boilers as feed water. If the condensing water is foul or contains matter that would be injurious to the boilers, a surface condenser should be used. If the water of condensation is not to be used in the boilers, the jet system may be employed. Another type of condenser that is being successfully used with steam turbines is the Bulkley injector condenser.

Among the steam turbines that were on exhibition at the St. Louis exposition in 1904 the Westinghouse-



Fig. 201. Spindle or Rotor, Allis Chalmers Steam Turbine.
The rings which carry the blades are pressed on.

Parsons and the General Electric Curtis turbines were each equipped with Worthington surface condensers, fitted with improved auxiliary apparatus consisting of dry vacuum pumps, either horizontal of the well-known Worthington type, or rotative motor-driven, a hot well pump, and a pump for disposing of the condensed steam from the exhaust system. The two latter pumps were of the Worthington centrifugal type. The Hamilton-Holzwarth turbine was equipped with a Smith-Vaile surface condenser, fitted with a duplex double-acting air pump, a compound condensing circulating pump, and a rotative dry vacuum pump, motor-driven. The vacuum maintained was high, 28 to 28.5 in.

As an instance of the great gain in economy effected by the use of the condenser in connection with the steam turbine, a 750 K. W. Westinghouse-Parsons turbine, using steam of 150 lbs. pressure not superheated and exhausting into a vacuum of 28 in., showed a steam consumption of 13.77 lbs. per B. H. P. per hour, while the same machine operating non-condensing consumed 28.26 lbs. of steam per B. H. P. hour. Practically the same percentage in economy effected by condensing the exhaust, applies to the other types of steam turbines.

With reference to the relative cost of operating the several auxiliaries necessary to a complete condensing outfit, the highest authorities on the subject place the power consumption of these auxiliaries at from 2 to 7 per cent of the total turbine output of power. A portion of this is regained by the use of an open heater for the feed water, into which the exhaust steam from the auxiliaries may pass, thus heating the feed water and returning a part of the heat to the boilers.

A prime requisite to the maintenance of high vacuum,

with the resultant economy in the operation of the condensing apparatus, is that all entrained air must be excluded from the condenser. There are various ways in which it is possible for air to find its way into the condensing system. For instance, there may be an improperly packed gland, or there may be slight leaks in the piping, or the air may be introduced with the condensing water. This air should be removed before it reaches the condenser, and it may be accomplished by means of the "dry" air pump.

This dry air pump is different from the ordinary air pump that is used in connection with most condensing systems. The dry air pump handles no water, the cylinder being lubricated with oil in the same manner as the steam cylinder. The clearances also are made as small as possible. These pumps are built either in one or two stages.

A barometric or a jet condenser may be used, or a surface condenser. The latter type lessens the danger of entrained air, besides rendering it possible to return the condensed steam, which is pure distilled water, to the boilers along with the feed water, a thing very much to be desired in localities where the water used for feeding the boilers is impregnated with carbonate of lime or other scale-forming ingredients.

In comparing the efficiency of the reciprocating engine and the steam turbine it is not to be inferred that reciprocating engines would not give better results at high vacuum than they do at the usual rate of 25 to 26 in., but to reach and maintain the higher vacuum of 28 to 28.5 in. with the reciprocating engine would necessitate much larger sizes of the low pressure cylinder, as also the valves and exhaust pipes, in order to handle the

greatly increased volume of steam at the low pressure demanded by high vacuum.

The steam turbine expands its working steam to within 1 in. of the vacuum existing in the condenser, that is, if there is a vacuum of 28 in. in the condenser there will be 27 in. of vacuum in the exhaust end of the turbine cylinder. On the other hand, there is usually a difference of 4 or 5 (2 to 2.5 lbs.) between the mean back pressure in the cylinder of a reciprocating condensing engine and the absolute back pressure in the condenser.

It therefore appears that the gain in economy per inch increase of vacuum above 25 in. is much larger with the turbine than it is with the reciprocating engine. Mr. J. R. Bibbins estimates this gain to be as follows: between 25 and 28 in. there is a gain of $3\frac{1}{2}$ to 4 per cent per inch of increase, and at 28 in. 5 per cent. These results have been obtained by means of exhaustive tests conducted by Mr. Bibbins. Other high authorities on the steam turbine all agree as to the great advantages to be derived by incurring the extra expense of erecting a condensing plant that is capable of maintaining the high vacuum necessary to high efficiency.

Another method by which the steam consumption of the turbine may be materially decreased and a great gain in economy effected is by superheating the steam. The amount of superheat usually specified is 100° , and the apparatus employed for producing it may be easily mounted in the path of the waste gases. The steam may thus be superheated without extra cost in fuel, and an increase of 8 to 10 per cent in economy effected. The independent superheater requires extra fuel and labor, and the gain in this case is doubtful, but there can be no

question as to the wisdom of utilizing the waste flue gases for superheating the steam.

As previously stated, the steam turbine is peculiarly adapted for the use of highly superheated steam and high vacuum, and in these two particulars it excels the reciprocating engine. At the present time many large plants are equipped with turbine engines that are giving the best of results, and the outlook for the future employment of this type of power producer is certainly very promising.

QUESTIONS.

233. What is implied in the proper adjustment of the valves of an engine?

Ans. The valve or valves must be so adjusted that the following named functions will each occur at the proper moment during one stroke of the piston: first, admission; second, cut-off; third, release; fourth, exhaust closure.

234. What two factors are necessary in order that the valve may be able to perform these functions?

Ans. The valve must have lap, and lead.

235. How does the valve receive its motion?

Ans. Through the medium of the eccentric.

236. What is the first step in valve setting?

Ans. To place the engine on the dead center.

237. Which direction should the engine be moved, when placing it on the dead center?

Ans. In the direction in which it is to run.

238. Why should the engine be moved in that direction?

Ans. On account of the lost motion of the moving parts, which would permit of considerable movement of the wheel, before any movement of the cross-head would take place. This would result in an error in taking measurements.

239. What is the next step in valve setting?

Ans. To properly adjust the length of the eccentric rod?

240. What is next?

Ans. To place the valve in its central position on its seat and mark it.

241. How is the correct travel of the valve ascertained?

Ans. Connect the valve rod to the eccentric rod, and turn the eccentric around on the shaft to the limit of its throw in one direction and measure the distance the valve has traveled from its central position. Next turn the eccentric to its extreme throw in the opposite direction. This should move the valve the same distance from its central position, in that direction.

242. If the valve does not travel equal distances each way from its central position what must be done?

Ans. Its travel must be equalized by making the necessary changes.

243. What is the next important move in valve adjustment?

Ans. To set the eccentric at its proper location on the shaft.

244. How is the proper lead given to the valve?

Ans. Through the angular advance of the eccentric.

245. How does a shaft governor control the speed of the engine?

Ans. By varying the angular advance of the eccentric, which with this class of governors is free to move across the shaft.

246. Explain the chief points of difference between the action of the reciprocating steam engine, and the steam turbine?

Ans. The piston of the reciprocating engine is driven back and forth by the static expansive force of the steam; while in the steam turbine, not only is this static expansive force made to do work, but the velocity of the steam in expanding from a high, to a low pressure is also utilized in turning the rotor of the turbine.

247. What other important factors enter into the operation of a steam turbine?

Ans. The principles of reaction and impulse.

248. Name several of the more important advantages that the turbine has over the reciprocating engine?

Ans. First, highly superheated steam of a high initial pressure may be used in the turbine. Second, a larger proportion of the heat in the steam may be converted into work with the turbine. Third, there is much less friction with the turbine.

249. What is the most economical method of disposing of the exhaust steam from a turbine?

Ans. By allowing it to pass into a condenser.

250. Will the turbine expand the steam to as low a pressure as the reciprocating engine will?

Ans. Yes, and even lower.

251. What type of condensing apparatus is necessary with the steam turbine?

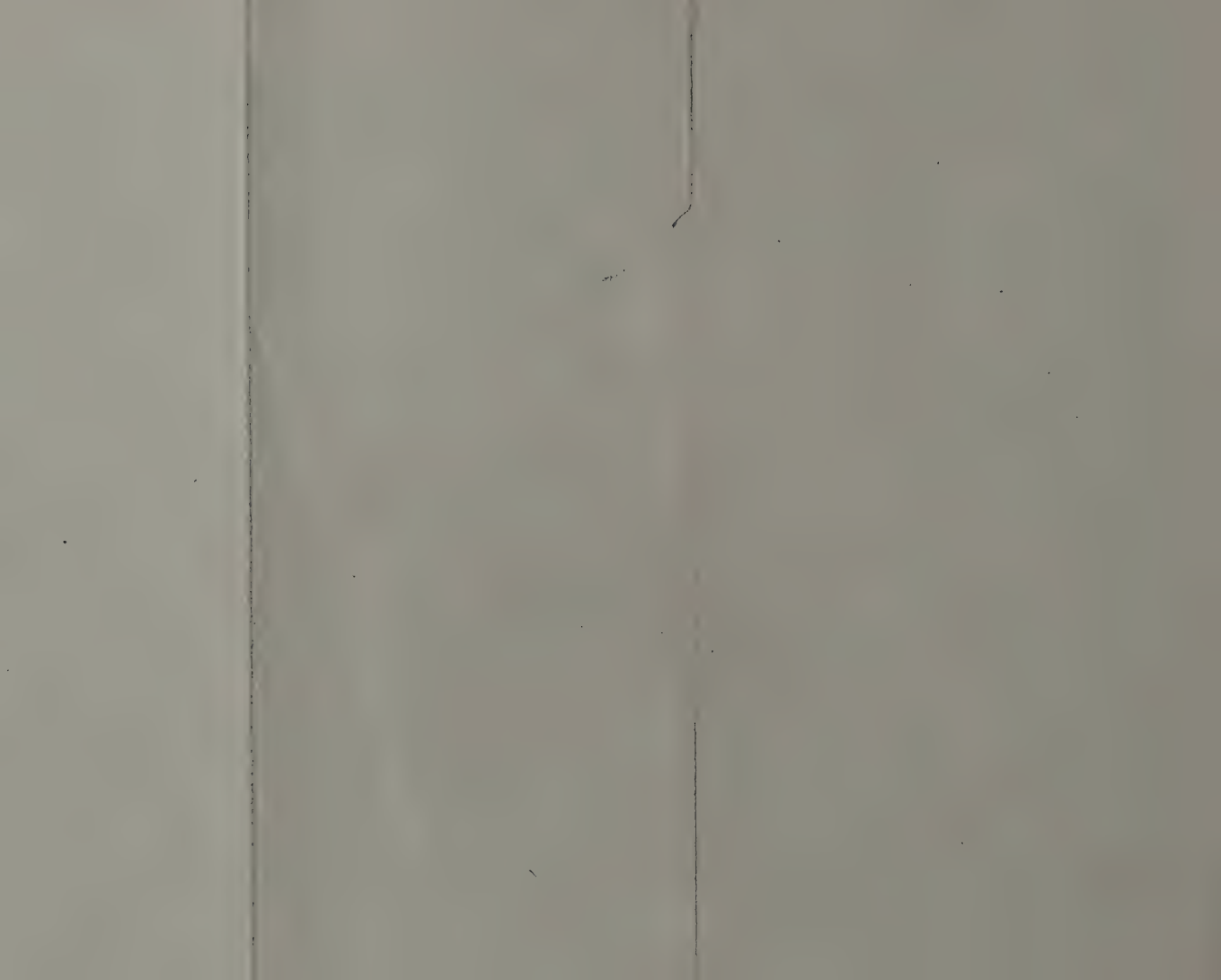
Ans. The same kind that is used on reciprocating engines.

252. How low will a well regulated turbine allow the steam to expand?

Ans. To within one inch of the vacuum existing in the condenser.



FIGURE 202



THE GAS ENGINE.

The gas engine differs structurally from the steam engine in two particulars, it is much more ponderous than a steam engine of equal output and has usually a much heavier crank-shaft. The difference in total weight must not be laid to the higher mean pressures exerted in the cylinder, for, in engines of a certain power, the mean pressure in the gas-engine cylinder will be higher than that in the steam-engine cylinder, so that for a given power per stroke the gas engine may have the smaller cylinder. But the four-stroke-cycle gas engine has only one working stroke in four. For a given power, therefore, its cylinder area must be four times the area of the steam-engine cylinder, per pound of mean effective pressure. Since this latter will be as 3 to 2, approximately, the actual cylinder area will be as 2.66 to 1, or the diameter ratio will be 1.63 to 1.

Approximately the gas engine appears fully 50 per cent the larger when its piston speed is the same as that of the steam engine. This is a very strong inducement to the designer to produce an engine which shall do work in both ends of the cylinder. Such a design, however, necessitates a piston-rod stuffing-box, gland and cooling devices. It involves double the number of explosions per minute in the cylinder and it renders possible a reduction of the cylinder ratio, as compared with a steam engine, to 1.33 to 1. Regarding single acting, or double acting gas engines, the difference in stress on the crank shaft must be taken into account. A single-acting en-

gine produces a torsion in the shaft which is reversed on the next stroke when the shaft is pushing back the piston. Reversing stresses are about 50 per cent. more destructive than stresses in one direction only. The crank-shaft strength ratio is thus not simply 4:1, as between a four-stroke and a one-stroke method of working, but it is as 6:1, or diametrically as 1.8:1, the strength varying as the cube of the diameter. This is why gas-engine crank shafts are so very large and this point must be of great importance on the score of cost. Indeed the four-stroke engine embodies a large amount of material which does very little work during a large proportion of its working hours.

The gas engine is a prime mover which derives its power or energy from the heat generated by the combustion within its cylinder, of a mixture of gas and air in the proper proportion to form an explosive. The combustion of this charge of gas and air is occasioned under a close or heavy compression, a result of the inward movement of the piston after the charge is admitted and all valves closed. The result of igniting this mixture under the heavy compression is what is commonly called an explosion, which is nothing more than a quick burning or rapid combustion of the mixture. This sudden explosion causes a high degree of heat within the cylinder behind the piston, and, the resultant high initial pressure against the piston drives it forward, and, through the medium of connecting rod and crank, motion is imparted to the main engine shaft. The original gas engines, and a majority of the smaller sizes of today, operate upon the Beau de Rochas cycle or four stroke cycle, sometimes termed the Otto cycle, meaning that an engine completes a cycle in four acts, defined as

follows: (1) Induction—During an outstroke of the piston, air and gas in suitable proportions are drawn into the cylinder. (2) Compression—The following in-stroke compresses the combustible mixture into the clearance space. (3) Explosion—Ignition of the compressed charge causes a rapid rise of pressure and subsequent expansion of products. (4) Expulsion—The expanded gases are expelled by the returning piston. In this type of gas engine, two revolutions of the crank shaft are necessary in order to complete one cycle.

Many small engines and some of those of largest power are designed upon the 2-stroke cycle, which is as follows: (1) Compression of the charge. (2) Ignition, explosion and expansion, and at the end of the stroke the exhaust products are expelled and the cylinder filled by a mixture of gas and air under pressure. In the two cycle engine, two compression chambers are necessary due to the fact that in this type of gas engine consisting of two cylinders, either side by side, or tandem, the charge of gas and air is being received in one cylinder, while the previous charge in the other cylinder is being compressed, preparatory to explosion. A two cycle engine thus explodes a charge, and receives an impulse at each revolution. It is important to admit only pure air and gas into engine cylinders. Dust and grit, or tarry matters cause rapid wear of interior surfaces. Care is also necessary to insure the induction of cold charges, in order that maximum density of gas and air may be obtained.

Induction. The charge of gas and air in definite proportions is drawn into the cylinder by the suction of the engine piston, and the velocity of entry is in direct proportion to the piston speed. The air valve is usually op-

ened before the gas valve, but inasmuch as there is no suction created until the opening of the air valve, some makers set the valves so that the gas valve is approaching its maximum lift by the time that the air valve has commenced to open, thus ensuring a well mixed charge. Usually, however, the settings are arranged so that the first portion of the induced charge is of air only, then air and gas, and finally air with the small quantity of gas swept in by the still moving current of air from the passages connecting the gas and air valve seats. The cams operating the valves are carefully designed to permit maximum lift with swift but gradual opening and closing, to accord with the induced velocity set up by the linear speed of the piston at each point of the stroke. The air valve, governing the entry of the entire charge, is opened well in advance of the inner dead center of the engine and is kept from closing until after the outer dead center, so that full effect of the momentum imparted to the entering gases at the highest rate of piston speed can be utilized without restriction, it being possible by such means to obtain a better filled cylinder.

Compression. Modern practice in gas engine design aims at securing the economical advantages coincident with high compression pressures, but the limit of allowable maximum compression pressure depends upon the relative proportion of hydro-carbon gases, and hydrogen contained in the mixture or charge admitted to the cylinder. Hydrogen will ignite at a much lower temperature than the other constituents, and owing to the additional heat during compression, it becomes necessary to so design the relative volumes of piston displacement, and clearance, that self ignition is practically impossible. With blast furnace gas containing only about two per

cent of hydrogen, compression pressures of 200 lbs. per sq. in. and over may be safely used, and with producer gas, 150 to 200 lbs. are common and safe pressures, but with illuminating gases the maximum is placed at 120 lbs. per sq. in. unless special precautions are taken to insure efficient cooling and cleaning of the cylinders. This is effected by the injection of water or cold air through the clearance spaces and valve ports during the charging stroke, or by pressure during compression.

Ignition. The increase of compression pressures and the use of poor gases for power purposes has brought electrical ignition devices into common use. Hot tubes of porcelain or hecnun are still used for engines designed to suit illuminating gas, but the impossibility of quickly adjusting the instant of explosion when running, the rapid deterioration of timing valves, and cost of renewals have emphasized the superior advantages of electrical firing. While with petrol engines the current from a primary or secondary battery is utilized with intensifying coil, and jump sparking plug, the usual method adopted for gas engines is that of a positive break by mechanical separation of two electrodes through which current is passing from a magneto machine. In a magneto the lines of force flowing between opposite poles of a permanent magnet of great strength is alternately deflected from, and passed through an interposed armature by means of a shield or deflector operated by suitable mechanism from the half speed shaft. Upon maximum rapidity of the armature cutting the magnetic lines of force a strong current is induced in the windings and passes through a circuit formed by an insulated wire connected to a fixed, well-insulated electrode through the second and movable electrode in elec-

trical contact with the engine frame and through this to the armature. It is of course very necessary to time the mechanism making the "break" so that it synchronizes with the period of the most powerful induced current in the armature. Most of the difficulties encountered with magnetos have been owing to the slipping of the actuating mechanism from the coned seating on the armature spindle, but once the correct setting has been noted and marked, attendants have found that very little other attention is necessary.

CHARACTERISTICS OF THE EXPLOSIVE MIXTURE.*

"Theoretically, ignition must be effected early enough and be so efficient that the whole of the power charge is ignited when the piston reaches the inner dead center position. Thus the condition of maximum heat development will occur with the minimum cooling surface and in a space which is specially designed to withstand high temperatures and pressures. Purity, calorific value, temperature and compression of the mixture, as well as the position and efficiency of the sparking apparatus, the form of combustion chamber and other conditions, will cause inflammation to spread faster or slower, which phenomenon becomes quite clearly visible on the indicator card, provided the latter be taken on a drum with continuous travel.

In several types of large modern gas engines the point of mixing the gas and air is rather superficially treated, while Reichenbach, who certainly deserves consideration as an authority on the subject, puts very much emphasis on this feature in all his designs from the earliest down

*Franz Ehrich Junge,

to the very latest. It is interesting to examine what has actually been done to clarify this question, and which view the serious student of the gas problem is justified in holding.

Gas and air properly mixed in chemical proportions, so that just sufficient oxygen is present in the combination to ensure perfect combustion will give the highest temperature of explosion which it is possible to obtain. Above and below this ideal condition there is a wide range of inflammability wherein more or less oxygen in the form of air may be mixed with the gas, than is necessary for its chemical combustion, so that a mixture of such composition will yet ignite but will burn at a slower rate of flame propagation and, consequently, will not develop the maximum temperature corresponding to its calorific value. If with a certain gas there be mixed about 4.7 times the amount of air that is necessary to establish the condition of chemical balance, the mixture will be that which is theoretically best suited for adoption in gas-engine practice. Theory and practice often differ, and so it is found advantageous to employ in actual practice far more air in the internal combustion process than is theoretically required. The reasons are threefold: To reduce temperatures all round, to prevent premature explosions which might be provoked by the high heat of compression, and to supply to the gas, even when poorly mixed with the air, always sufficient oxygen for combustion, and consequently to reduce the loss of unburnt gases leaving the exhaust to a minimum.

If one examines by thermodynamic calculation the combustion efficiency of lean mixtures under whatever cyclic conditions they may be transformed into work, it will be found that maximum economy is attained by com-

pressing the weakest mixture to the highest possible degree, but here again one is confronted by an upper limit which is rigidly drawn by the lack of inflammability of such mixtures. Desire for thermal excellence of the working process forces us to approach this upper limit as much as possible, but the decreasing calorific value of the power charge per unit of contents and the decreasing capacity of the engine keeps the actual practice far below this extreme ideal. In average practice it is customary, at normal loads and with lean gases, to work with a surplus of air of from 30 to 40 per cent over what is theoretically required; with gases of high heat value, even more air is provided, so that the dynamic medium in the engine cylinder possesses a calorific value of from 44 to 62 B. t. u. per cubic foot."

Proper timing of ignition devices has a direct result upon the economical working of the engine. If the mechanism is set too early on the compression stroke, combustion of the charge occurs at or before the inner dead center of the engine, resulting in violent shocks, excessive strains upon the piston, connecting rod, and bearings, and involving great waste of power. If too late, the piston has commenced to accelerate under the influence of momentum stored up in the flywheel, so that the explosive force follows the piston without attaining its maximum thrusting effort, and with some loss of compression pressure due to the re-expansion of the charge before ignition is effected.

The speed of flame propagation varies with the percentage of hydrogen contained in combustible mixtures, and it is convenient for means to be provided to adjust the instant of ignition to suit varying qualities of gas. For this reason many makers fit a device permitting an

attendant to set the "break" at the most suitable instant, and there is no doubt that with such facilities a careful man can thus obtain good results even with very variable mixtures such as often occur with poor gases generated by producers. Some leading manufacturers, however, realize that, by carelessness or neglect, the provision of such means of adjustment is liable to misuse, and they prefer to arrange two firing points only—one very late for starting up at slow speeds, and another for normal speeds, the variations in inflammability of charges being deemed of less importance than the variations of the average skill and intelligence of attendants.

Explosion and Expansion. The mean pressure upon the area of the piston throughout the stroke is, of course, of great importance and directly affects the power given out from the engine. High initial explosion pressures *per se* do not create the most powerful efforts behind the piston, neither are low terminal expansion pressures indicative of maximum economy.

Exhaust. Before the expansion or power stroke is fully completed the exhaust valve commences to open—usually when the piston has still to travel one-tenth of its stroke. The terminal expansion pressures then are about 25 to 30 lbs. above atmosphere, and the velocity with which the burnt product leaves the cylinder due to such pressure imparts considerable momentum to the column of escaping gases, thus helping to effect their thorough evacuation. With long exhaust pipes not unduly restricted, the energy of the moving column of gases is taken advantage of.

Relative Efficiency of Power Gases. Apart from the calorific values of one cubic foot of the various power-gases when burnt in sufficient air to support complete

combustion, it is necessary to differentiate them in terms of calorific value per cubic foot of gas and air mixture when only just sufficient air is present. For gas engines, however, it is necessary to dilute the gas still further in order to control ignition as, varying with the percentage of hydrogen present, theoretical mixtures are impossible, owing to risks of pre-ignition and violent shocks caused by the rapidity of the propagation of flame. Experience has determined that while hydrogen is of the greatest value in obtaining good results, yet it is important that its volume should be only about 7 per cent in gas engine mixtures.

Indicated Horse Power. This is a computation based upon the mean effective pressure developed at each explosion and is usually calculated from the same formula used in connection with steam engines: $I. H. P. = \frac{P L A N}{33,000}$ where P = mean effective pressure; L = length of stroke (ft.); A = area of cylinder; N = number of explosions per minute. This formula does not discriminate between mechanical friction and losses in "fluid" friction. To get accurate results it is necessary to obtain the mean effective pressure after measuring the indicator diagrams recorded during both "power" and "cut-out" cycles as also "compression" and "suction" cards.

Economy of Gas Engines. As fuel is ordinarily used, at present, for light, heat and power, the losses are so great that, of the total calorific value of the coal, less than 5 per cent on an average is converted into useful work, while the largest and best power stations utilize only about 10 per cent.

With gas-engine-driven units, however, fuel economy is the distinguishing characteristic, resulting in a deliv-

ery at the engine shaft of not less than 16 to 20 per cent of the energy contained in the fuel.

Notably is this true of lignite coal, immense quantities of which are to be found in various parts of the country. The calorific value of such coal averages only about 8,000 B. T. U. per pound as fired; but, when utilized in a gas producer and gas engine, it is possible to develop a brake horse power on less than 2 pounds of lignite.

Owing to the difficulty of securing, with this fuel, proper combustion under a steam boiler, the gas producer offers practically the only means of using it on a commercial basis.

The same is true of slack coal, bone coal, etc.

A recent bulletin of the United States Geological survey calls attention to the possibilities of the producer gas plant, as above indicated, and states that lignite beds underlying from 20,000,000 to 30,000,000 acres of public land, heretofore supposed to be practically useless, are now shown to have a large value for power development. This is of particular importance to the West, making possible a great industrial development there.

Producers now made will also successfully gasify nearly all grades of bituminous coal, as well as anthracite or coke.

ADDITIONAL ADVANTAGES OF GAS ENGINES.

The standby losses during a period of idleness are practically negligible, and are included in the fuel estimate per brake horse power as given above, leakage losses are reduced and the smoke nuisance is abolished.

In localities where the water supply is scarce, or of

poor quality, a gas power plant offers additional advantages, and, by the use of cooling towers, the water consumption can be reduced to a very small quantity.

GAS PRODUCERS.

At present three distinct types of gas producer are offered to the power user. They are the suction producer, the steam-pressure producer, and the induced down-draft producer.

THE SUCTION PRODUCER.

In this type the fuel is fed into the generator from a hopper at the top. Ashes and clinkers are removed from the bottom, and air is usually admitted below the grate, first passing through economizers, where it is heated and passed over a body of hot water to absorb the necessary moisture. In some makes of producer the air is admitted direct from the engine room, and a small, regulated amount of water is fed into a space prepared around the grate, where it is evaporated and is carried as steam along with the air up into the fuel bed.

In this type of producer, coke or anthracite coal can only be used and not even these fuels in the very small sizes. It is not easy to note the condition of the fire, as the generator cannot be opened at the top without admitting air and causing a poor mixture of gas; the only thing to do in this emergency is to feed in more coal to stop the chimney holes in the fire-bed, or quickly insert a poker bar and thoroughly tamp the fuel. The latter is the better way, even though it has to be done blindly.

In operating this type of producer, trials and tribulations may be many and varied, depending largely upon

how the producer is made and the basis of its horse power rating. A suction gas producer rated at more than $12\frac{1}{2}$ pounds of coal per square foot of grate area, or area of fuel-bed cross-section, is very apt to be too small, and a producer so small for the power it has to develop that it must be driven to furnish sufficient gas will immediately develop clinker troubles, variable gas troubles or excess CO_2 . If an attempt is made to correct clinker troubles with an over-supply of steam, an excess of hydrogen will result, with its attendant engine difficulties of back-firing, or premature explosions.

Even with a producer of the proper size, the regulation of the volume of steam or water to the volume of air must be closely watched. Too frequent raking of the fire will waste good fuel, and induce draft holes through the fuel bed. Too much poking from the top will pack the fire, and necessitate an increased vacuum. Too fine a fuel will produce the same troubles, and any coal that fuses easily will not do for this type of producer. Anthracite running high in slate mixture tends to run high in sulphur, and high sulphur with slate makes bad clinkers at any time, and if the fire is forced at all will soon necessitate a shut-down to clean out. The producer should be of ample size—10 pounds of coal per square foot of internal area—and rated on $11\frac{1}{4}$ pounds of coal per horse power hour.

It is not good practice to use a suction gas-producer plant of over 150 horse power when the engine has to draw the gas from the producer by the vacuum in the cylinder. Sizes larger than this should be equipped with exhaust fans which will relieve the engine of this work, the exhausters being driven by motors, or other auxiliary power.

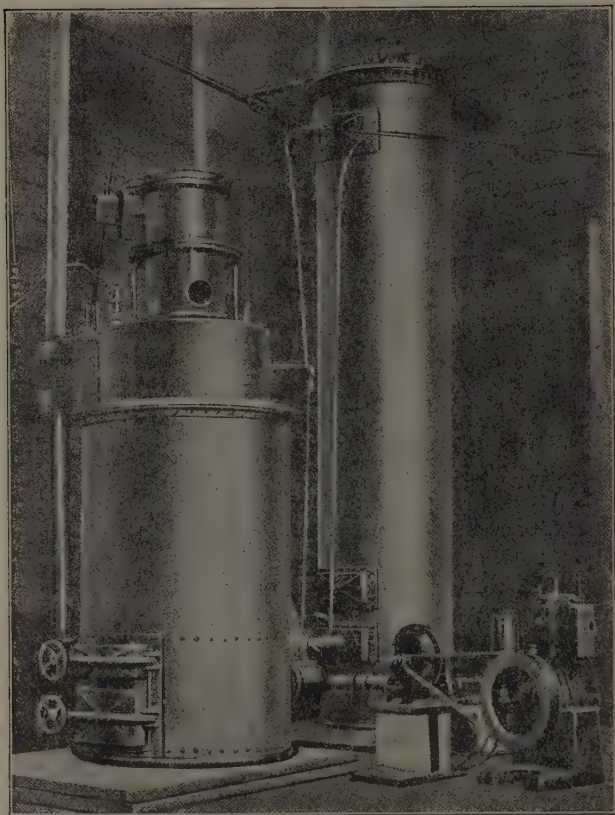


Fig. 203. Monahan Suction Producer.

THE MONAHAN SUCTION PRODUCER.

This producer, a full view of which is shown in Figure 203, is of the suction type, and consists of the usual generator, scrubber, and equalizing tank. The vaporizer for supplying steam to the fuel bed is an upper exten-

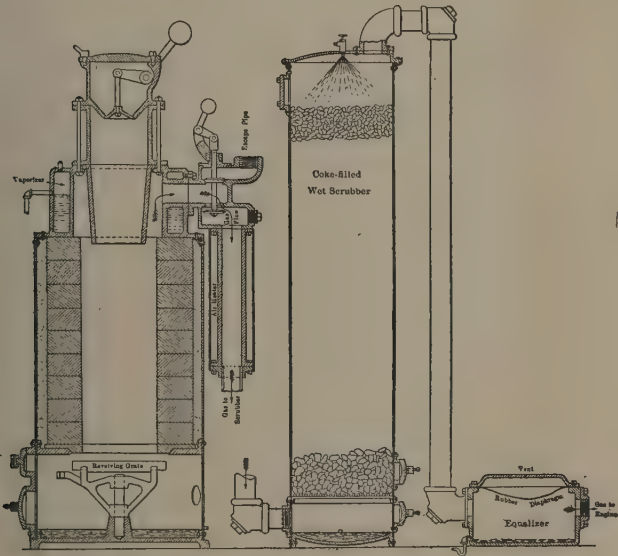


Fig. 204. Sectional View of Monahan Producer.

sion of the generator, but is located so that the hot gases from the fuel bed do not impinge squarely on the bottom of the vaporizer. This is clearly shown in the sectional view, Figure 204, in which the revolving grate, and air heater are also shown. The Scrubber is of the wet, coke-

filled type, and the equalizing tank is a simple drum within an equalizing chamber formed in its cover, and separated from the drum by a rubber diaphragm.

A small vent in the cover allows air to pass in or out slowly, forming a sort of brake on the fluctuations of pressure within the drum. The regulator controlling the admission of steam to the fuel bed is shown in section in

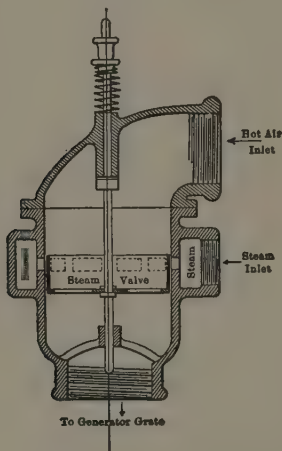


Fig. 205. Steam Regulator.

Figure 205. The outlet at the bottom is connected to the ash pit of the generator. The upper intake admits air only, and the intake near the middle admits steam. When running light the suction is insufficient to pull down the valve, and air alone passes through the fire. As the load increases, the increasing suction gradually pulls down the valve (which is a piston valve), until the

steam ports are uncovered to an extent depending upon the load. This arrangement prevents the chilling of the fire with steam at very light loads, and graduates the supply for heavier loads.

THE STEAM-PRESSURE PRODUCER.

This type has an upward draft, the air being drawn in around a steam jet through a K rting nozzle of the Bunsen type. Anthracite and coke are the only kinds of fuel available, unless tar extractors, and other expensive mechanical auxiliaries are provided to clean the gas. When the producer is equipped with such cleaning apparatus, bituminous coals or fuels containing volatile hydrocarbons may be used, but as these are condensed and washed out of the gas, the thermal efficiency of the producer is reduced to the extent of the loss of the heat units contained in the extracted hydrocarbons, which are the richest part of the fuel.

With a pressure producer it is necessary to have gas-storage capacity, so that gas-holders must be provided regardless of the kind of fuel used, and these must hold enough gas to run the engine while the fire is being poked, and the ashes removed. Coal is fed through a tightly closing hopper on top of the generator, and ashes are removed from the bottom when the generator is not in operation. It is almost impossible to poke or bar the fire while the producer is running, as any outlet for this purpose will be flooded with burning gas escaping under whatever pressure the steam jet is maintaining at the time.

THE INDUCED DOWN-DRAFT PRODUCER.

In the down-draft producer the gas is drawn down through the fire by an exhaustor or fan, and forced by the exhaustor through the main to the point of use. There is probably more horse-power of these producers in use than in all of the others put together, but they are mostly of large size and the plants only number about one-fourth of the total.

Essentially these are bituminous-coal producers. They are operated with an open top, where the fire is seen by the operator, and any blow-holes or passages in the fire are easily closed by the use of the poker or tamping bar, and fresh fuel is fed as necessary. The volatile hydrocarbons of the fuel, being distilled at the top of the fuel bed, mix with the in-drawn air and steam, and pass down through the bed of incandescent carbon, where they combine with the other gases and leave at the bottom of the producer as a fixed non-condensable gas. This combination of gases then passes directly into the bottom of a vertical tubular boiler and out at the top, thence into the bottom of the wet scrubber, where the outlet is under water to form a seal and prevent the gas from returning to the producer. From the top of the wet scrubber the gas passes to the exhaustor, and is forced through the dry scrubber to the gas-holder.

The boiler, which is a part of the producer installation, supplies a large part of the steam necessary for the producer and also the amount necessary to run the engine driving the exhaustor. This steam is made from the heat given up by the gas in its passage through the boiler, and all heat that is not absorbed by the water is

delivered up to the wet scrubber. Once a week these producers have to be entirely cooled down to be cleaned, and as the steam pressure in the boiler is down at this time, an auxiliary boiler has to be provided to start up again. Some time during the week, especially toward the last days, the fuel beds become so clogged with the accumulation of ashes and clinkers that water-gas runs have to be made every few moments; the load on the engine driving the exhauster increases, and both these conditions so increase the demand for steam that the auxiliary boiler has to be brought into use.

For continuous 24-hour service with this type of producer it is necessary to have a spare unit, in order that it can take the place of the one that has been in service for a week. A single spare unit in an installation of a large number of units does not add a very large percentage to the original investment, but a spare unit to a single outfit nearly doubles the cost. The following timely suggestions regarding gas engine practice are presented by the Gas Power Section of the A. S. M. E.:

“Engine Efficiency. Should be expressed in terms of effective heat value until a combined gas-vapor cycle comes into use. For the present, let us not confound a definite engine efficiency by introducing the indefinite factor of latent heat of water vapor. Engine efficiencies should be given for full, to half load at least.”

“Producer Capacity. The producer should be rated upon its ability to gasify coal. It would be more accurate to rate on B.t.u. of standard gas, but this is impracticable. Should the builder desire to rate on a special coal, he might insert a clause limiting some of the constituents. In specifying sizes, a maximum as well as a minimum screen should be mentioned. A mixture of

many sizes packs the producer as badly as a very small fuel. As a usual thing the flexibility of the producer will more than meet the overload possibilities of the engine.

Producer Efficiency. Can only be specified in terms of B.t.u. output, involving volumetric measurement,



Fig. 206. Allis-Chalmers Four-Cycle, Double-Acting, Tandem Gas Engine Direct-Connected to an Allis-Chalmers Continuous Current Generator—Natural Gas Used.

which it is usually impossible to determine except by calibration of the engine. As we are dependent upon the engine as a gas meter, we must be consistent and determine the efficiency of the producer in like terms, that is, the ratio between heat output in standard gas, and heat input in fuel for the fire.

Producer Regulation. An important point is the property of the producer as regards the regulation of heat value of the gas and its pressure as delivered to the engine. Quality regulation is covered by the engine-capacity clause "with gas of not less than so many B.t.u. heat value per cubic foot."

Hydrogen Content. This may be expressed as a percentage by volume of the gas, a percentage by volume of combustible in the gas, a percentage of the heat value of the gas per mixture, or a percentage by volume of the mixture. The last appears to be the most explanatory. The first conveys no impression of the commercial value of the gas. The second is better in this respect. The third presents widely varying values."

Figure 206 presents a view of a four cycle, double acting tandem gas engine as built by Allis-Chalmers Company of Milwaukee, Wis. This engine is using natural gas. Fig. 207 shows a four cycle double acting, twin-tandem gas engine by the same company.

The distinctive features of the gas engines built by Allis-Chalmers Co., or those which appeal most strongly to engineers who have seen them in service, are the extreme simplicity of design, the solidity of construction and the quiet operation. Maximum overloads are handled as easily, and with the same freedom from vibration that characterize their operation under normal conditions; the engines turn their centers as quietly as a slow-running Corliss machine and with apparent indifference to the rapid changes in load which they are often called upon to sustain.

While the engines are, as a whole, exceptionally rigid and heavy, the weight is concentrated in the frame cylinders and tie pieces in the direct line of stresses to which



Fig. 207. Allis-Chalmers, Four-Cycle, Double-Acting Twin-Tandem Gas Engines, Each of 2,000 H. P. Capacity, Driving Allis-Chalmers Alternating Current Generators in the Power House of the Milwaukee-Northern Railway, Port Washington, Wis.

an engine of this type is subjected. In the frame is illustrated the principal difference between European and American design. This frame is designed for a side crank, in place of the double throw crank which represents the standard practice abroad. The stresses transmitted to the frame in a side crank engine are very great, but, even in the largest sized gas engines, they are no greater than Allis-Chalmers Company has for many years successfully provided for in steam engine practice.

The jaw, which is subjected to peculiarly severe stress, is made in a form to insure maximum strength of the casting, and is further strengthened by two steel tie bolts carried above the shaft, which are made of sufficient size to carry their proportion of the load without appreciable elongation. This construction eliminates entirely any bending stresses in the frame at this point.

The engine frames for the 2,500 K. W. units weigh approximately 90 tons each, and one-half of each frame is buried in the foundation, in order to raise the floor line to a point which will make the slides on the valve gear readily accessible.

The pistons and rods are water-cooled, water being introduced at the center and flowing forward to a discharge in the frame for the front piston, and backward to a discharge in the tail guide for the rear piston, each piston having its separate supply. For dismantling or for cleansing, the rod is made in two parts joined at the central slide, the rear half going out at the back of the engine and the other half going out through the frame, which is made open at the top for convenience.

THE VALVE GEAR.

On twin tandems the valve gear is located between the engines, concentrating it in such a way as to make it very convenient for the operating engineer.

This gear is of Allis-Chalmers Co.'s stratification type and the engine operates with constant compression, thus tending to insure smooth running under the highly variable loads to which it is subjected. The inlet gear is extremely simple, consisting of a main inlet valve of the single beat poppet type, eccentric operated, thus insuring long life and quiet running. The mixture of the air and gas is thoroughly effected before entering the cylinder by means of a patented annular mixing chamber located under the main inlet bonnet; the design and operation of this device is such that, at the instant of closing of the main inlet valve, there is practically no explosive mixture left outside the cylinder. The gas valve is of the double beat poppet type, controlled by a variable lift rolling lever operated by a single link connection to the main inlet, the lift of the valve and consequently the amount of gas admitted and the time of admission being regulated by the governors. The exhaust gear is of the single beat poppet valve type, eccentric operated, and is in this respect a duplicate of the main inlet gear. A feature of this engine is the location of the exhaust bonnet with its valve at the bottom of the cylinder, where all the dirt is removed by the action of the exhaust gases, and the provision of a substantial jack to lower the entire exhaust mechanism out of place to allow inspection and regrinding of the valve, which also serves to swing the valve chamber, with the valve and its operating mechanism complete, out to one side

where it can be reached by the crane hoist. The removal of one pin, either in the inlet, or exhaust mechanism, is all that is necessary to allow the removal of either the inlet or exhaust bonnets, with their valves and entire operating apparatus, without disturbing any adjustment whatever.

The igniters are electrically controlled and so arranged that the time of ignition may be regulated by a single hand wheel. Direct current at 80 volts is used in the ignition system. Duplicate independent igniters are provided at each end of the cylinder to insure prompt firing of low heat value gases and also to avoid the danger of shut down due to short circuit.

The air starting device consists of a small poppet inlet air valve at each end of each cylinder, operated by the layshaft. Air is admitted to each cylinder in turn at what would be the working stroke. As the high compression carried prevents the engine from stopping on the dead center, this arrangement insures the prompt starting of even a tandem engine without the use of a barring gear. These engines being twin tandem will, of course, start from any position.

LUBRICATION.

All wearing surfaces, including the main bearings, slides, crank and cross-head pins, are arranged for a continuous oiling system and the cylinders are lubricated by carefully timed admission of the cylinder oil, sight-feed oil pumps being used.

The engines shown in Figure 207 are of the following dimensions: Each engine has four cylinders 32 inches in diameter by 42 inches stroke, and operates at 107 revolutions per minute. Each unit is rated at 1,000

K. W. but both engines and generators were designed for large overload capacities.

The engines shown in Figure 202 are said to have the largest cylinder diameter of any gas engine yet built in the United States, the dimensions being 44 inches diameter by 54 inches stroke.

WESTINGHOUSE GAS ENGINES.

The Western New York and Pennsylvania Traction Company operating in the vicinity of Olean, N. Y., have recently equipped their central power station with two tandem, double acting Westinghouse gas engines, and one twin tandem double acting Snow engine, each driving a 300 K. W. General Electric three phase alternator, at 150 R. P. M.

The Westinghouse engines in this plant have cylinders 22 inches diameter by 30 inches stroke.

Figure 208 is a vertical section through one of the Westinghouse cylinders, showing the gas and air distribution, water jacket, and also shows the valve gear. The inlet and exhaust valves are located at opposite ends of the vertical cylinder diameter, as shown in Fig. 208. Both valves, at each end of a cylinder, are operated by a single eccentric through wipers and rocker-arms; this construction is also shown in Fig. 208. The exhaust valve is of the mushroom type, hollow, with a tube extending up the stem into the center of the head for discharging the cooling water; the water is introduced through the annular passage between the outlet tube and the wall of the valve-stem. The valve cage sets into a circular housing projecting downward from the cylinder.

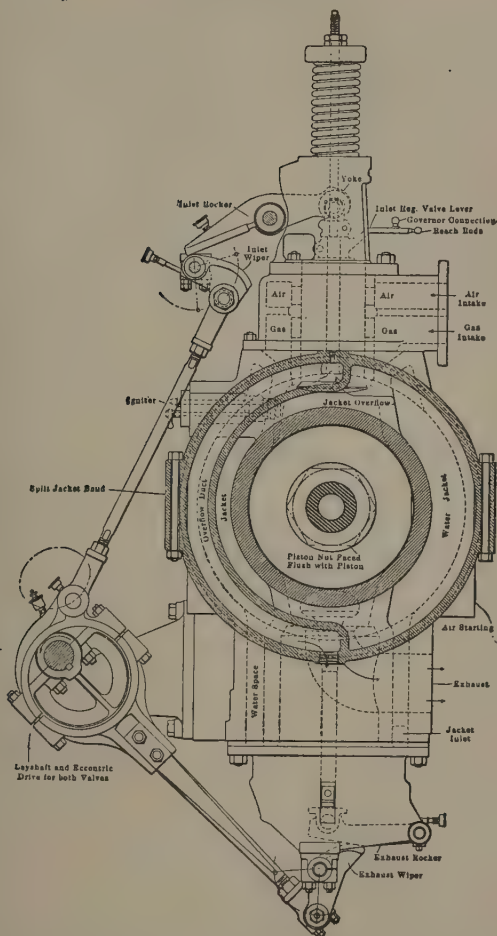


Fig. 208. Valve Gear, Westinghouse Gas Engine.

The main inlet valve is of the simple disk type, and is opened and released by the eccentric and wiper levers always at the same points of the piston travel. There is mounted on the valve-stem, however, a cylindrical valve which controls the quantity of air and gas admitted, this being under the control of the governor. It is not shown in Fig. 208, but the connection for the governor rod is shown. The cylindrical valve fits closely in the bore of the valve cage and has ports in its wall which correspond to the ports leading into the cage from the air and gas passages. When the admission valve is on its seat, the ports in the cylindrical valve are above those in the cage wall, and the latter are therefore closed, preventing gas from backing up into the air passage. When the inlet valve is depressed by the valve-gear, the cylindrical valve goes with it, and its ports then come into horizontal alignment with the air and gas ports in the wall. The governor controls the quantity of air and gas admitted by rotating the cylindrical valve on the disk-valve stem; in the full-load position the ports are all in exact alignment when the valve is depressed, and at lesser loads the valve is twisted around by the governor so as to shut off part of the port opening. The four cylindrical valves of one side of the engine are all connected together by reach-rods, so that the governor adjusts all four valves simultaneously and alike. The two sets of regulating gear are connected by double reach-rods, so that there is no lost motion between the two sides. The pistons are centered in the cylinders by adjustments at the three crossheads. Each piston is equipped with four sectional packing rings; the sections of each ring are joined by brass keepers and set out by flat steel springs. The piston-rods are hollow, of course,

to admit and discharge cooling water to and from the pistons, and they are turned without any camber.

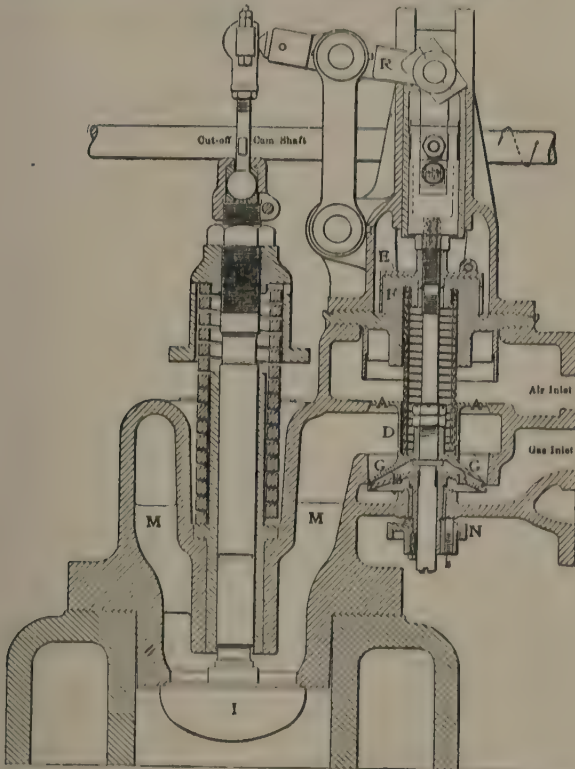


Fig. 209. Sectional Elevation of Inlet and Cut-off Valves and Gear. Snow Gas Engine.

SNOW GAS ENGINE.

The Snow twin unit has cylinders 16 inches in diameter by 30 inches stroke. The cranks are set at 90 degrees

apart, thus giving four impulses to the shaft per revolution. The combustion chamber is on the side, with the inlet valves in the top, and the exhaust valve in the bottom of each combustion chamber.

The engine speed is regulated by cut-off valves which, under control of the governor, shut off both the gas and air supply sooner or later in the suction strokes, according to the speed changes. The arrangement of inlet valves for one end of a cylinder is shown by Fig. 209. The air and gas pass to the mixing chamber *M* through separate ports, shown closed by the valve disks *A* and *G*, respectively. From the mixing chamber the mixture is admitted to the cylinder by the main inlet valve *I* at the beginning of the suction stroke; at the point in the stroke determined by the governor, the cut-off valve *A G* is released and allowed to close under the influence of its spring. The baffling disk *B* is adjustable so as to obtain the desired proportion of gas to air, the adjustment being made by means of the nurlled head *N*, which is locked in the proper position by the set-screw *s*. The shank of the baffling disk serves also as a guide for the lower end of the stem of the cut-off valve. The valve disks *A* and *G* are connected by a short barrel *D*, the whole being a single casting. The gas valve *G* is provided with a tapered seat, and the valve stem is adjusted in the block at its upper end until both disks seat simultaneously.

The main inlet valve *I* is opened and closed always at the beginning, and termination of the suction stroke, by the inlet rocker-arm (Fig. 210), and its stem is linked to a short rocker-arm *R*, Fig. 209, to the other end of which is pivoted a block arranged to slide vertically in a guide. To this block is hung the pivoted latch *L*, shown

in Fig. 210, the end of which normally engages a dog on the block which is screwed on the upper end of the stem of the cut-off valve. When the main inlet valve is opened, the latch *L* lifts the disks *A* and *G* of the cut-off valve. At the proper point of the suction stroke, the cam *C*, Fig. 210, engages a lug and draws the drag-

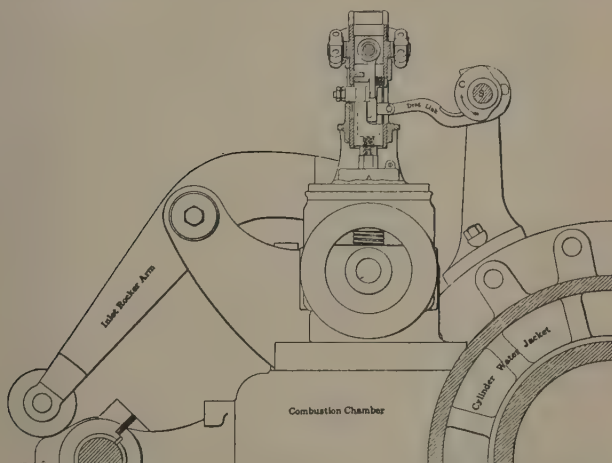


Fig. 210. End Elevation of Inlet and Cut-off Mechanism.
Snow Gas Engine.

link over, thereby pulling out the latch *L* and allowing the cut-off valve to drop. The drag-link is pivotally attached to a lug on the latch *L*, and its other end is curved around the cut-off shaft *S*, the upper leg of the bend resting on the journal box and holding the link in place as it slides back and forth. Before the succeeding suction stroke begins, the cam *C* has turned to the "low" side and the latch *L* is thrown into engagement with the

valve-stem dog by a small helical spring. When the cut-off valve drops, it is cushioned by the inverted cup *E*, Fig. 209, acting as a dash-pot, the plug *F* constituting the plunger. The cut-off cam-shaft *S* rotates continuously at one-half the crank-shaft speed, and its angular position with respect to that of the crank-shaft is adjusted by the governor through the well-known "floating" bevel gear.

THE DU BOIS TANDEM GAS ENGINE.

The Du Bois Iron Works, Du Bois, Penn., has developed and is now building a line of single-acting tandem gas engines which embody several interesting features. Fig. 211 is a view of one of these engines, from which it will be evident that the design conforms to the standard European practice of locating the inlet valves in the top, the exhaust valves in the bottom, and the valve-gear shaft alongside of the cylinders. Another characteristic European feature is the use of center-crank construction. Beyond these few points, however, the design cannot be said to follow strictly any classified practice.

The longitudinal section, Fig. 212, gives an excellent idea of the internal construction of the engine, and shows clearly the unusual method employed for packing the piston-rod hole in the rear end of the front cylinder. Instead of providing a stationary packing cage, and having the rod slide through rings contained therein, the packing rings are put on the rod, like those on the piston, and they slide back and forth with the rod in a sleeve formed in the head of the front cylinder and a rearward extension of it. It might seem at first glance that this arrangement would entail an unduly long engine structure, but the fact is that even if stationary packing rings

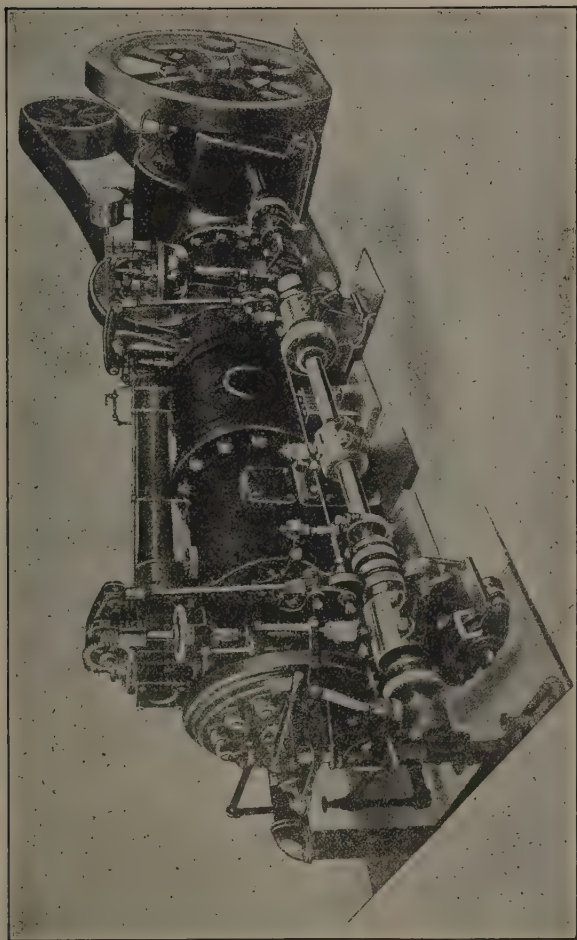


Fig. 211. Du Bois Tandem Gas Engine.

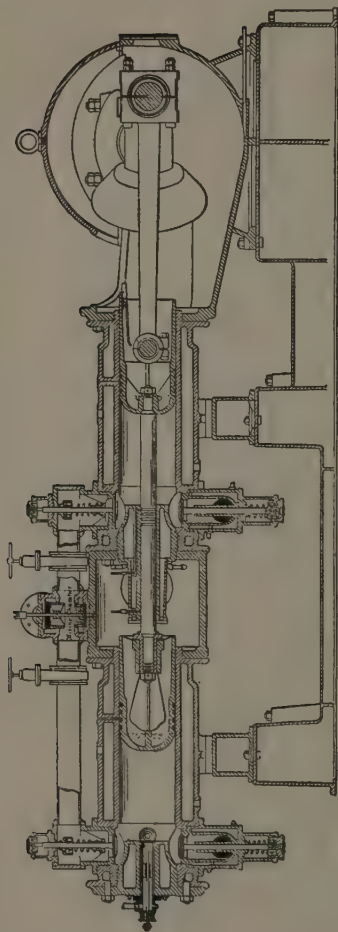


Fig. 212. Longitudinal Section of Du Bois Tandem Gas Engine.

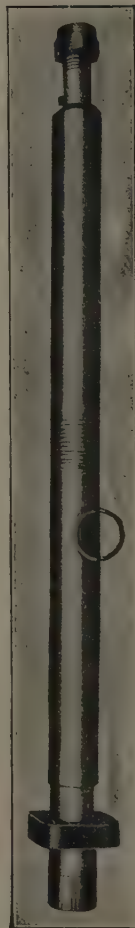


Fig. 213. Piston Rod of Du Bois Gas Engine.

were mounted in a housing in the cylinder head, the engine could not be shortened up without sacrificing accessibility to the rear piston and ease of dismounting the front cylinder head. Fig. 213 is a view of the piston rod with its packing rings and the flanged sleeve to which the rear piston is bolted.

The pistons are of the trunk type, and are built with convex heads, in order to obtain the requisite strength with a moderate weight of metal.

The construction of the water jackets, cylinder heads, connecting rod and crank case is so clearly shown by Fig. 212 as to require no verbal description. The crank shaft, crank cheeks and pin are all cast in one piece of steel; the balancing weights are bolted on.

VALVES, VALVE GEAR AND GOVERNOR.

Simple, flat poppet valves of forged steel, with beveled seats, are used throughout the engine, and the need for cooling the exhaust valves in larger sizes is obviated by the use of auxiliary exhaust ports uncovered by the pistons at the end of the forward stroke. These ports consist of a series of round holes through a rib connecting the water-jacket wall and the cylinder barrel, and they are drilled, instead of being cored, in order to obtain absolute accuracy in dimension and location. The inlet and exhaust valves of each cylinder are opened by a cam, two push rods and four rocker arms, as indicated in Fig. 214. Rollers are provided, of course, to take the thrust of the cam and to deliver the motion of the rocker arms to the valve stems.

One mixing valve serves both cylinders, as indicated in Fig. 212, where the mixing valve is shown immedi-

ately above the rear end of the piston-rod packing sleeve. Fig. 215 shows more of the details. The valve stem car-

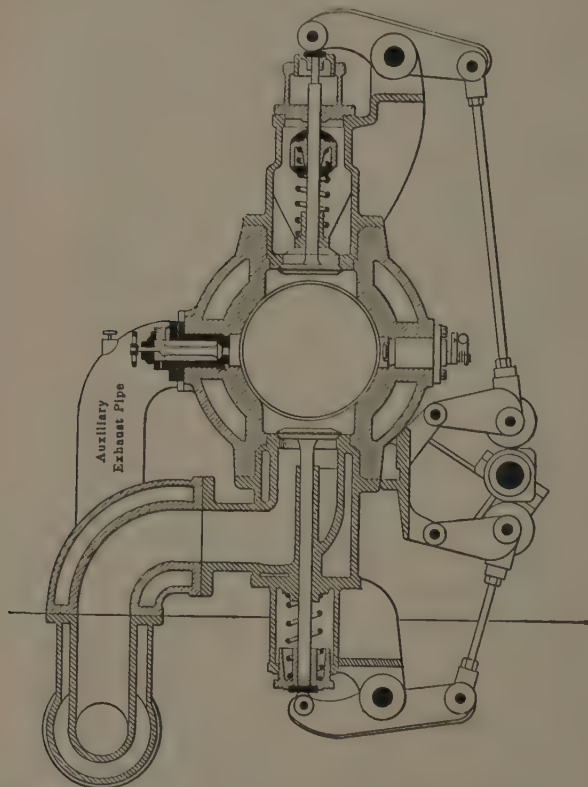


Fig. 214. Cross Section Through Cylinder and Valve Chambers, Du Bois Gas Engine.

ries two pistons, the upper one controlling the air supply by varying the space between its upper edge and the top

of the cage, and the lower one varying the gas supply by means of ports in its wall and corresponding ports in the wall of its cage. The proportion of gas to air is adjusted manually by means of the butterfly valves in the supply passages, except in such cases as require variation

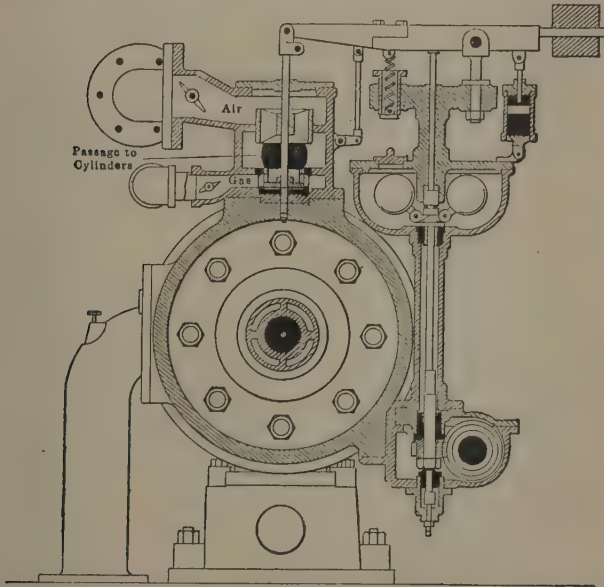


Fig. 215. Cross Section Through Cylinder, Mixing-Valve Chamber and Governor, Du Bois Gas Engine.

of the mixture proportions simultaneously with variation in the quantity of mixture admitted. For such conditions, the lever which raises and lowers the mixing valve is also linked to the butterfly valves, the linkage being adjusted so that the mixture is made richer as the load

decreases (and the compression is reduced), and poorer as the load increases. This automatic mixture control is

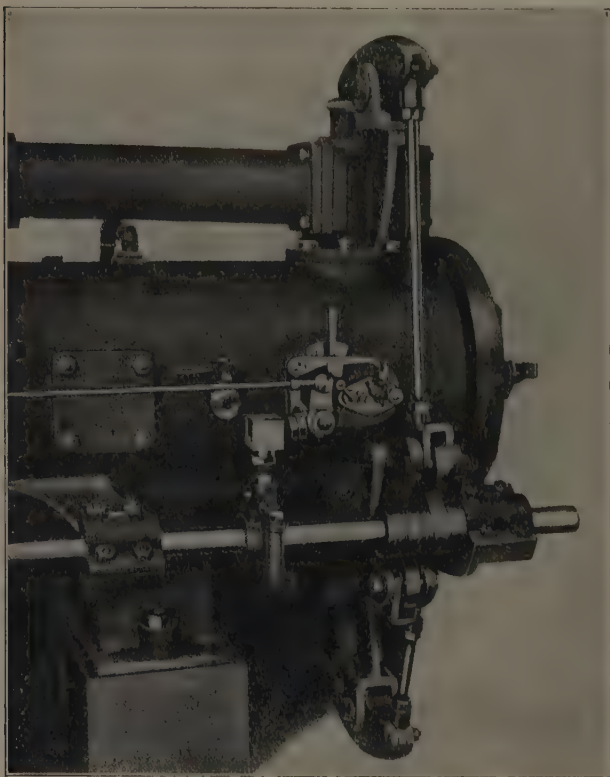


Fig. 216. Du Bois Igniter Mechanism.

not necessary except when running on very lean gases. The mechanism is adjusted usually to give the best mix-

ture proportions at full rated load, but a wide range of adjustment is practicable.

The governor is of the flyball type, but differs essentially from the common construction, as Fig. 215 clearly shows. The spindle is driven through spiral gears from the cam shaft, and the balls and sliding member are inclosed in a stationary housing, as shown in Fig. 215. The governor gears are located between the forward cam and the main shaft; consequently, the angular velocity of the governor is not disturbed or made irregular by the tensional yielding of the cam shaft to the stresses imposed by the cams and valve mechanism. The cam shaft is driven from the crank shaft through spiral gears running in an oil bath.

IGNITION.

Igniters of the mechanical make-and-break class are used. The reciprocating mechanism which trips each igniter is driven by an eccentric on the cam shaft, as shown by Fig. 216. The individual igniter is timed by means of the vertical handle near the cylinder; this raises or lowers the horizontal finger with the bent end which trips the igniter, and thereby alters the point of ignition. The reach rod leading from the igniter mechanism to the left extends to the other igniter, and adjustment of this rod to the right or left retards or advances the timing of both igniters simultaneously. The levers to which the ends of this rod are pivoted are fastened to the ends of sleeves which are eccentric to the rocker studs, and the igniter rockers are mounted on these sleeves; turning the sleeves on the studs alters the relation between the rockers and the igniter triggers and thereby changes the tim-

ing. A handle at the middle of the reach rod serves for manipulating it, and a clamp holds it wherever it is set.

The engine is arranged so as to be oiled by a central gravity-feed system. Since the piston trunks are more than full-stroke length, they always cover the oil holes in the cylinder walls; timed lubrication is therefore unnecessary.

The engine is equipped with a valve in the head of the rear cylinder for starting with compressed air, and the valve disk is located at the inner face of the cylinder head, so that when it is closed no pocket is formed in the combustion-chamber wall. This construction is shown in Fig. 212. The indicator openings are provided with similar valves, as shown in Fig. 214. This drawing also shows the unusual feature of a water-jacketed exhaust pipe with which every Du Bois engine is equipped. Another unusual feature, although not original on this engine, is the injection of cooling water into the exhaust pipe; this cools the exhaust gases so suddenly that a muffler is not required. The water jacket around the exhaust pipe is chiefly for the purpose of obviating the exposure of dangerously hot surfaces where attendants are likely to come in contact with them, although the water jacket also cools the exhaust gases considerably.

The compression pressure is about 140 pounds absolute for natural gas, and 180 pounds for producer gas, and the engine can be changed from the one to the other compression in a few minutes.

A NEW CROSSLEY GAS ENGINE

Figure 217 shows a gas engine recently built by the well-known English firm of Crossley Brothers, Ltd. (Manchester, Eng.), for driving a cotton mill in Japan.

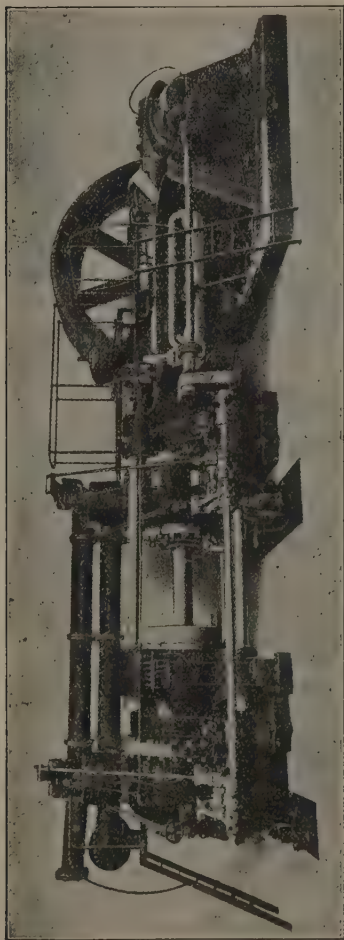


Fig. 217. A New Crossley Gas Engine.

This engine is a considerable departure from the opposed-cylinder type which has become so familiar. It is a tandem single-acting engine, having composite inlet valves, and an ingenious pneumatic governor control of the admitted charge. The cylinders are 38 inches bore and the stroke is 39 inches; the normal speed is 105 revolutions per minute. The engine is rated at 650 horse power, with a maximum ability of 700 horse power, on producer gas.

The exhaust and inlet valves are both new designs. The exhaust valves are of the balanced type, working horizontally, and are very readily accessible, as they can be drawn out at floor-level for examination and cleaning.

The special feature of the admission valves is the arrangement for regulating the gas supply in accordance with the requirements of the engine as the load on it varies. There are for each cylinder two valves—a main valve and a gas valve. The former is positively driven in the ordinary manner, and is of the usual mushroom pattern. The gas valve is mounted, with a sliding fit, on the spindle of the main valve. The two valves—that is to say, the main and the gas valve—always close together; but the main valve is positively opened by the usual cam and push-rod and the gas valve is opened by the partial vacuum which is established below it on the suction stroke of the engine. The time at which it opens and consequently the quantity of gas drawn into the cylinder are controlled by the governor through a vacuum pot. The gas valve is rigidly connected with a piston which is slidable in a small cylinder to which the engine governor admits more or less air through a balanced piston valve. At full load, this piston valve is held wide open, and air can readily enter the vacuum cylin-

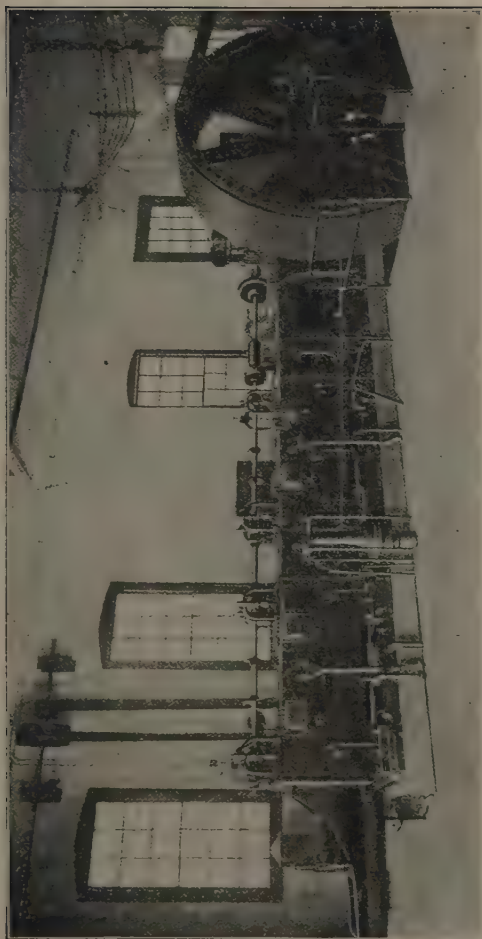


Fig. 218. Operating Side of Riverside Gas Engine at the Watson-Stillman Plant, at Aldine, N. J.

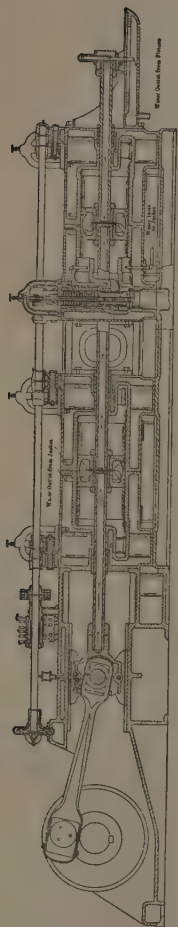


Fig. 219. Longitudinal Section of Tandem Engine at Aldine.

der; the gas valve is, accordingly, free to move as soon as the slightest vacuum is established below it. As a consequence the engine receives a full charge of gas. As the load on the engine falls off, the governor partially closes the small piston valve, and the gas valve cannot then open so freely, its motion being checked by the fact that as soon as it begins to move there is a reduction of pressure in the vacuum cylinder behind the piston to which the gas valve is connected. It therefore opens later, and less gas enters the working cylinder. This method of regulation has been found to be very sensitive and reliable in operation and it obviously dispenses with the often complicated trig-gears frequently used for the same purpose.

The valve-gear shaft does not extend all the way to the crank-shaft, but is geared to a separate length of lay-shafting by two-to-one gears at the forward end of the forward cylinder. The governor is driven from the full-speed lay-shaft, and its rate of rotation is, therefore, not disturbed by the periodic retarding influence of the valve push-rods nearly so much as it would be if driven from the half-speed shaft.

Ignition is effected by means of oscillating low-tension

magnetos and make-and-break igniters mechanically operated. There are two igniters in each cylinder, and each igniter is operated by a distinct set of rock-shafts and links, and supplied by an individual magneto.

A small pump driven from the valve-gear shaft delivers lubricating oil through sight feeds to the different parts of the engine.

The forward cylinder is bolted to the main frame and fitted to a circular countersunk seat; the rear cylinder is held to the main frame by four tie-rods, as indicated clearly in the engraving.

QUESTIONS.

253. In what respect does the gas engine differ from the steam engine structurally?

Ans. It is a much more ponderous machine than a steam engine of equal output and usually requires a much heavier crank shaft.

254. Why should this be?

Ans. Because the ordinary four-stroke-cycle, gas engine has only one working stroke in four, and requires four times as much cylinder area for a given amount of work, as would a steam engine for the same work.

255. Define the difference between a single acting four stroke cycle and a double acting or two stroke cycle gas engine in their operation.

Ans. In the four stroke engine two revolutions of the crank are required for one cycle. In the double acting or two stroke, the cycle is completed in one revolution of the crank.

256. Why are gas engine crank shafts made larger in proportion than those of steam engines?

Ans. In order that they may withstand the increased torsional strains.

257. What causes the pressure behind the piston of the gas engine?

Ans. The combustion within the cylinder of a charge of gas and air properly mixed to form an explosive, and admitted at the proper moment.

258. When is this proper moment?

Ans. When the piston is at the end of its instroke ready to start outward.

259. Define the stages of a four cycle engine?

Ans. First, induction; during an out stroke of the piston, air and gas are drawn into the cylinder in the proper proportions. Second, compression; on the return in stroke the piston compresses this combustible mixture into the clearance space. Third, explosion; ignition of the compressed charge causes a rapid rise of pressure and subsequent expansion of products. Fourth, expulsion; the expanded gases are expelled by the returning piston.

260. Define the stages of a two cycle gas engine?

Ans. First, compression of the charge. Second, ignition, explosion, and expansion, and at the end of the stroke the expanded products are expelled, and the cylinder filled by another charge of air and gas under pressure.

261. How many compression chambers are needed for the two cycle gas engine?

Ans. Two; for the reason that this type of gas engine requires two cylinders, either side by side, or tandem, and the charge of gas and air is being received in one cylinder, while the previous charge in the other cylinder is being compressed preparatory for explosion.

262. What precaution should be observed regarding the admission of the air and gas?

Ans. The air should be pure and free from dust, and the gas should not contain tarry matters if it can be avoided.

263. How are the induction valves usually set?

Ans. So that the first portion of the charge is air only, then air and gas, and finally air with a small quantity of gas.

264. How is the air valve controlling the entry of the entire charge adjusted?

Ans. It is set to open well in advance of the inner dead center of the engine, and is kept from closing until after the outer dead center.

265. Why is this valve so set?

Ans. In order that the full effect of the momentum imparted to entering gases at the highest rate of piston speed may be utilized.

266. Upon what does the allowable compression pressure depend?

Ans. Upon the relative proportions of hydro-carbon gases, and hydrogen contained in the mixture.

267. What per cent of hydrogen is considered within the limits of safety?

Ans. Not over 7 per cent.

268. What are the usual compression pressures carried with blast furnace gas?

Ans. 200 lbs. per sq. in.

269. What pressure may be safely carried when producer gas is used?

Ans. From 150 to 200 lbs. per sq. in.

270. If illuminating gas is used, what is the maximum safe pressure?

Ans. 120 lbs. per sq. in.

271. How is the cylinder cooled and cleaned?

Ans. By the injection of water or cold air through the clearance spaces, and valve ports during the charging stroke, or by pressure during compression.

272. What other methods are available for cooling the cylinder and piston rod?

Ans. By means of a water jacket that surrounds the cylinder. The piston rod may be hollow and water circulated through it.

273. How is the charge of gas and air ignited?

Ans. Formerly by hot tubes of porcelain or hecnum, which are still used to some extent, but at the present day electrical ignition devices are used principally.

274. What is one of the most important features connected with ignition?

Ans. To see that ignition occurs at the proper moment.

275. At what point in the stroke of the piston should ignition occur?

Ans. This depends upon the quality of the gas used. With the maximum allowable percentage of hydrogen, ignition should not occur until after the piston has passed the inner dead center. Otherwise the result will be violent shocks, and strains upon the working parts.

276. Do high initial explosions create the most powerful efforts behind the piston?

Ans. They do not.

277. What are the usual terminal pressures for gas engines?

Ans. 25 to 30 lbs. above atmospheric pressure.

278. How is the horse power of a gas engine calculated?

Ans. Usually from the same formula used in connection with the steam engine, and the computation is based upon the mean effective pressure developed at each explosion.

279. What percentage of the total calorific value of the coal is usually converted into useful work with the steam engine?

Ans. From 5 to 10 per cent.

280. What percentage of the energy contained in the fuel is it possible to utilize with a modern gas-driven unit?

Ans. From 16 to 20 per cent.

281. How many types of apparatus are in use for the production of gas for power?

Ans. Three: the suction producer, the steam pressure producer, and the induced down draft producer.

282. What kind of fuel must be used in the suction producer?

Ans. Coke or anthracite.

283. What kind of fuel is available for the steam pressure producer?

Ans. Only coke or anthracite.

284. What kind of fuel is the induced down draft producer adapted for?

Ans. Bituminous coal.

285. How may gas engine efficiency be expressed?

Ans. In terms of heat value.

THE DYNAMO.

The dynamo is a machine for transforming mechanical energy into electrical energy—mechanical energy is required to operate the mechanism for changing field and armature relations, and this energy is absorbed by the dynamo, and electric energy is produced in its stead. The easiest way to comprehend the principles of the dynamo is to follow up its construction from the most simple type, to one of the more complicated forms. Dynamos are classified into two grand divisions, viz., alternating (A. C.) dynamos, and direct current (D. C.) dynamos. The A. C. dynamo produces a current that reverses its direction of flow periodically, in practice from twenty times and upward per second. The D. C. dynamo produces a current of unchanging direction.

The principal constituent parts of a dynamo are the armature, consisting of a core and windings, the field consisting also of core and windings, the collecting rings, or commutator and brushes. The armature and field vary in construction, their windings vary in system, and from these variations, many different varieties of dynamos are constructed.

Direction of the Induced E. M. F. Lay a magnet on a table with its N-end projecting over the edge. Take a copper wire and connect its two ends to a galvanometer or pressure meter; usually called a *Voltmeter*. Stretch a portion of this wire between the right and left hands, and move the wire rapidly down in front of the N-end of the magnet at right angles to lines of its action. There will

be induced in the wire a pressure or *Electro-motive force* which will send current through the wire from the right hand to the left. The galvanometer will give a deflection showing the flow of current.

This deflection is not a permanent one, the needle instantly dropping back to zero, proving that only a momentary current was produced.

If the experiment is tried moving the wire upwards the direction of the momentary current is reversed.

With a downward motion in front of a S-pole we get a left to right current.

Hence, starting with a certain polarity and direction of motion, changing one changes the direction of the induced current, while changing both the polarity and the direction of the motion does not reverse the current.

Now hold the wire a foot away from the magnet and directly opposite it, moving the wire up to the magnet and back again, keeping the distance from the wire to the floor the same, so that the motion shall be parallel to the flux. No. E. M. F. is generated in this case.

An electromagnet would do as well and probably better for the experiment as they are usually stronger than permanent magnets.

The value of the Induced E. M. F. depends upon the following:

1. The greater the strength of the magnet, the greater the E. M. F.
2. The more rapidly the wire is moved, the greater the induction.
3. The larger the number of turns in a coil of wire, the greater is the E. M. F. induced in it.

The induced E. M. F. therefore depends on the *flux*, the

speed of cutting this flux, and the number of wires cutting this flux.

A dynamo reduced to its simplest form is a coil of wire arranged so as to cut the magnet flux of an electro magnet, thus producing an induced electromotive force.

The current flows through the dynamo and the external circuit in *series*, hence the greater the current the larger must be the dynamo and the heavier the wires of the external circuit.

Since the E. M. F. (electromotive force) can be produced almost entirely by speed of the machine, the voltage at which the current is delivered does not affect the weight of the machine very much.

It is also evident that the faster a machine is driven the smaller the magnets can be, and yet the same E. M. F. be produced. This is why high speed generators weigh less than low speed machines. The usual 600 volt railway generator weighs 11, 13, or 16 pounds per ampere of current capacity.

The railway generator whose voltage is 600, and whose resistance is 0.025 of an ohm, would give a current which can be calculated by Ohms law.

$$\text{Amperes} = \frac{\text{E. M. F.}}{\text{Resistance of generator plus resistance of the external circuit.}}$$

If we allowed the external resistance to fall too low by attempting to operate too many locomotives at the same time, the current drawn would generate a great amount of heat while passing through the machine. The amount of this heat can be calculated by the rule.

Heat in Watts is equal to the Square of the current multiplied by the resistance of the generator.

If we build an 11-ton generator and load it too heavily, that is, put 4,000 amperes on it; it will have C^2R watts of heat or $16,000,000 \times 0.025$ equal to 40 K. W. (kilo watts) to dissipate. Now a machine like this has such a small surface that it cannot radiate 40 K. W. of heat to the air, whereas if a heavier machine had been built, one of 22 tons, it could have gotten rid of the heat. Moreover the copper wires of the 22-ton machine being larger than those of the light machine, the resistance they offer is less, and there will be less heat generated. Thus the heavier machine will run cooler than even the light machine properly loaded.

The danger in overheating of generators is the damage done to the cotton insulation of the wires, due to scorching, and the melting of the shellac varnish between the layers of mica. The mica itself stands the heat very nicely. Any overheating of the armature and the commutator is at once conducted to the bearings.

To prevent the generation and retention of too much heat it is customary to allow 700 C. M. (circular mils) of copper for each ampere of current, and to design the shape of the coils of wire so that they will have one square inch of surface for every two watts of heat to be gotten rid of.

The generator must be protected by fuses (which will melt) or by circuit breakers (automatic switches opening at a definite current) which will open the circuit, and prevent the further flow of current in case of an accidental low resistance or *short circuit*, as it is called, which would otherwise draw a very heavy current and *burn out* the dynamo. The armature is the part that suffers first.

In talking about dynamos or generators (the two names are used for the same machine) we use the words *E. M. F.*

and *voltage*. By the E. M. F. of a dynamo is meant the *total pressure generated by the armature*. As the current flows through the armature and field circuits of a railway generator it encounters the *internal resistance* of the generator, and pressure is lost according to the rule: drop in pressure is equal to the product of the current and the resistance. Hence the current as it flows out of the generator is at a reduced pressure; and this pressure at which current is delivered to the switch board, is called the terminal voltage or simply the *Voltage* of the machine. For the generator like above $\text{drop} = C \times R = 2000 \times 0.025 = 50$ volts.

If we intend to deliver power at 600 volts it will be necessary to design the speed and the magnetic flux to produce a pressure of about 650 volts.

If a voltmeter be applied to the brushes of a generator just ready to go into service, but not yet carrying any load, the reading obtained is called the E. M. F. When the switches are closed and the generator furnishes power to the line the reading of the voltmeter will drop and its reading is called the *Voltage* of the machine.

We often speak of current being drawn from the dynamo. The generator keeps a steady pressure on the line of 600 volts and the flow of current is regulated by the load in the following way:

One locomotive hauling a train at speed will have a resistance of 0.3 ohm and will draw 600 divided by 0.3 or 2,000 amperes.

It would be more accurate to say 2,000 amperes are allowed to flow. This locomotive is the only path for current to pass from the feeder (third rail) to the return (rails). When two locomotives are in use there are two paths between the feeder and the return. The conduc-

tivity between conductors has been doubled and the resistance halved. Double the previous current now flows.

In this way we get the current desired by lowering the resistance, and as the current passes it performs the work.

Two locomotives draw twice the current of one, hence their conductivity is double that of one, and their combined resistance is half of one.

The conductivity of three locomotives is treble, so their combined resistance or joint resistance is one-third that of one.

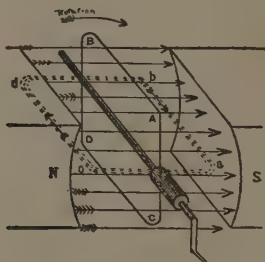


Fig. 220. Direction of Induced E. M. F. in Dynamo.

In the usual type of railroad generator, the revolving part is the armature and the stationary part the field magnets. Fig. 220 shows the elements of a dynamo, and as the loop is rotated there is an E. M. F. generated in it.

While the loop is coming into and passing away from the position ABCD, there is no E. M. F. generated, for the reason that the motion is parallel to the flux. As the loop moves towards the position ABCD by a uniform rotative speed it cuts the lines of force faster and faster, for first it cuts in an oblique manner but as the loop comes into the position of abcd the loop is moving straight across the flux. Hence the actual number of lines cut

per second is least at ABCD and gradually increasing becomes greatest at abed; thus producing an E. M. F. of varying value.

During this quarter of a revolution the wire BA has moved down in front of a S-pole inducing an E. M. F. tending to send current from B towards A. The other part of the loop CD has moved upward in front of a N-pole inducing an E. M. F. tending to send current from C towards D.

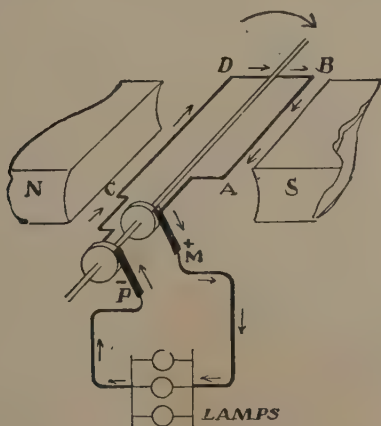


Fig. 221. The Simple Alternating Current Dynamo. Brush M Is Positive.

These results are in accordance with the experiments previously described.

This action is repeated during the next quarter of a revolution, and when finally the coil is in the position ABCD with AB at the bottom the pressure is again at zero.

The value of the E. M. F. has started at zero, risen to a maximum, and decreased to zero again. This gives a fluctuating current or pulsating current in the external circuit.

When AB rises in front of the N-pole the E. M. F. will be in the direction of from A to B, while before it was from B to A. During each revolution of the loop the current flows one way half the time, and then is reversed and flows the other way.

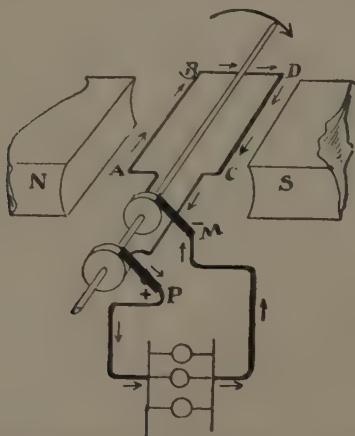


Fig. 222. The Simple Alternator, Shows Coil at One-half a Revolution from Fig. 221. Brush M is Now Negative.

This is what happens in the armatures of all dynamos whether alternating (A. C.) or direct (D. C.) current types.

In order to utilize the current flowing in the loop of Fig. 220 it is necessary to attach two collector rings as shown in Figs. 221 and 222, which gives us an A. C. gen-

erator or an *Alternator*. For a D. C. generator or simply *Generator*, the ends of the loop are connected to one ring split as shown in Fig. 223, whose halves are insulated from each other and from the shaft. Two brushes are placed as in Fig. 224. In this case the alternating E. M. F. will be *reversed* or *commuted* at the proper instant and there will be a one direction E. M. F. impressed on the external circuit. The split ring is called a *Commutator*.

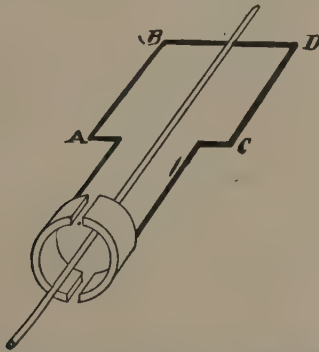


Fig. 223. An Armature Coil Connected to a Two-part Commutator, so as to Deliver Direct Current.

THE ALTERNATOR.

In Figs. 221 and 222 are shown two positions of the loop on the armature of an alternator. The collector rings are insulated from the shaft and each other by mica. The terminals of the loop are soldered or riveted (sometimes both) to the rings, and current is led to the external circuit containing the lamps by stationary strips of copper which form a sliding contact with the rings.

Referring to Fig. 221 it will be seen that during the

first half of the revolution of the loop ABCD, the direction of the E. M. F. in AB is from B to A, and in CD is from C to D.

The current flows from the brush M to the lamps so that M is positive.

Reference to Fig. 222 shows that the wire in front of the S-pole is still positive, but that it is now the wire CD instead of AB, so P is the positive brush for the second half of the revolution. There are two reversals of the current per revolution.

The number of *alternations per minute* is the speed in revolutions per minute multiplied by the number of poles.



Fig. 224. Cross Section of Simple Commutator. Black Represents Copper; White Space Is Mica Insulation.

The number of *cycles* is found by multiplying the speed in revolutions per second by the number of pairs of poles. The number of cycles is usually spoken of as the *Frequency* of the alternator.

The usual frequencies are for power 25, for motor circuits, and arc lamps 66, and for incandescent lighting 133.

THE DIRECT CURRENT GENERATOR.

In Fig. 225 is shown a loop and a two part commutator of a D. C. generator.

Since the wire AB is moving down past a S-pole, the current flows from B to A and out of the brush M, which

is called the positive brush. In wire CD the current flows from C to D, making P the negative brush.

After half a revolution the wire CD is over where AB was, and is now delivering current towards the external circuit instead of away from it; but *CD is now connected through its commutator bar to brush M instead of to P so that the brush M is still positive.* (See Fig. 226.)

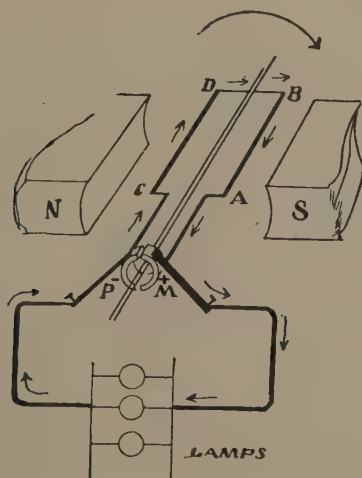


Fig. 225. Simple D. C. Generator. At This Instant the Brush M Is Positive.

This arrangement of commutator bars and brushes performs the duty of connecting the brush M to that part of the winding, and only that part which is moving down in front of a S-pole. As long as the wire AB moves up in front of a N-pole the commutator connects it to brush P, but as soon as it begins to move down in front of a S-pole

it is immediately disconnected from P. and a connection made with M.

To increase the E.M.F. The greater the field strength and the higher the speed the greater the E.M.F.

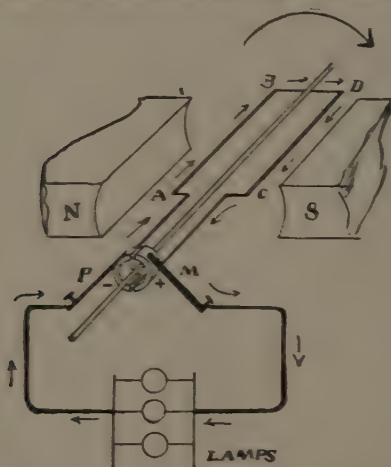


Fig. 226. Simple D. C. Generator. The Armature Has Made Half a Revolution, but Brush M Is Still Positive.

When the speed has been raised until the surface of the armature is traveling at the rate of 3,000 ft. a minute* no further increase is made, lest the bursting stresses become too great.

In order to further increase the E.M.F. more *turns* or *loops* of wire must be wound on the armature. A coil of

*This is called the Peripheral Speed of the armature and is calculated by this rule:

P. S. equals $3.1416 \times D \times R. P. M.$ where D is the diameter of armature in feet and R. P. M. is the revolutions of the armature per minute.

16 turns as in Fig. 227 will give an E.M.F. 16 times as great as a coil like Fig. 223. Reference to Fig. 228 will serve to make this plain.

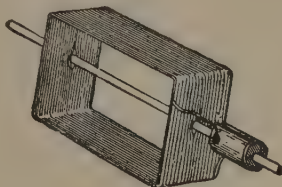


Fig. 227. A Single Coil Armature of Many Turns.

Suppose the direction of rotation to be the same as the hands of a watch when viewed from the commutator end of machine; then the E. M. F.'s induced in the successive

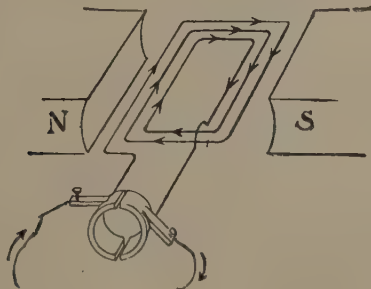


Fig. 228. An Armature Coil of Many Turns Showing How the Induced E. M. F. of Each Turn Adds Itself to That of Other Turns.

portions of the wire will be as shown by the arrows, and will add to each other impressing a high E.M.F. on the brushes. These turns of wire are said to be in series.

Any betterment of the magnetic conductivity of the frame of the machine will increase the E.M.F.; by pro-

ducing a greater flux per pound of copper on the field magnets. Hence the winding of the armature inductors (wires) on a core of very softest iron is an economic ne-

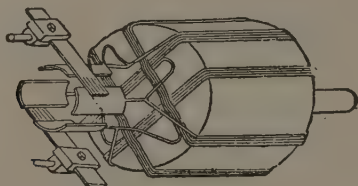


Fig. 229. Drum Winding on a Drum Core. Four Coils and Four Commutator Bars. For Direct Current.

cessity, resulting in either a higher E.M.F. or a reduction of the expense for copper in the field coils.

These cores are called *Drum cores* when the central hole is just large enough for the shaft and the insulation

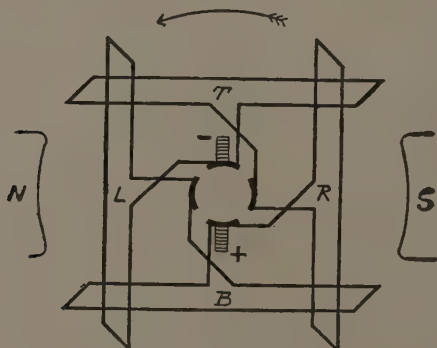


Fig. 230. Diagram of Fig. 229.

around it (Fig. 229); and are named *Ring cores* when the internal diameter of the ring is much larger than the shaft. (Fig. 231.) The armature in Fig. 232 has a ring

core, but the end plates being in position, the large hole is concealed.

These cores are built up of a great many punchings of soft iron from 15 to 40 mils thick, pickled so as to rust them a little. Every tenth one is varnished or tissue

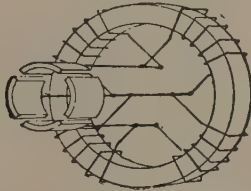


Fig. 231. Simple Gramme Ring Winding.

paper pasted on. The rust, varnish and paper are all insulators and when the punchings are assembled in a core, prevent currents called *Eddy currents* from flowing from one end of the armature to the other and heating it.

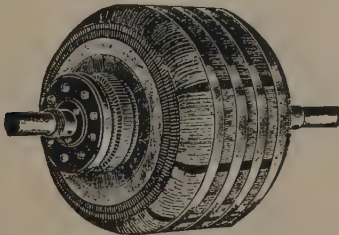


Fig. 232. Eight Section Eighty Coil Ring Winding on a Smooth Ring Core, with Eighty Bar Commutator. For Direct Current.

These cores are sometimes *smooth* but more frequently are *slotted* with the wires laid in the slots.

About 10 to 15% of the length of the core is insula-

tion, and about 50% of the surface is slots containing the inductors.

To get a Continuous E.M.F. While a single coil of many turns produces a high E.M.F., which by a two part commutator is always applied to the external circuit in the same direction, yet this coil passes through all the changes in voltage mentioned in connection with Fig. 220.

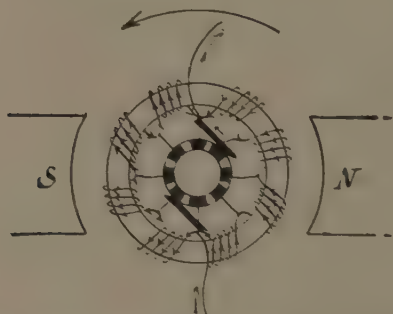


Fig. 233. Eight Coil Gramme Ring Winding, with Eight Part Commutator.

Fig. 231 shows the construction of the Gramme ring, so named from the inventor Gramme. The winding is on a ring core made up of soft iron punchings 25 mils thick.

The wires on the outer surface are *active*, having E.M.F. induced in them, and are called *armature inductors*. The rest of the wire is *dead wire* and only useful to complete the circuits between inductors.

Fig. 233 shows the same windings with eight coils and eight commutator bars. In Fig. 232 the armature as diagrammed in Fig. 233 is shown completed with its four bands. These bands are from 12 to 25 convolutions of

phosphor-bronze wire in sizes varying from No. 20 up to 14 laid on tightly over a mica insulation and sweated with solder all the way round.

In Fig. 233 it will be seen that the complete winding can be divided into two parts, one influenced by the N-pole, the other by the S-pole standing at the commutator end. The N-pole side moving upwards has its E.M.F. in direction from back to front of armature *through the inductors*; the S-pole side has E.M.F. in direction from back to front of armature *through the dead wire*.

In winding the armature the wire is laid on in a continuous spiral as shown. This makes the E.M.F. in each half of the armature in series, and allows the current to flow from one coil to another, except at the points where the N-half and S-half of the armature meet. Here the E.M.F.'s oppose and if wires were connected for an instant to the winding, as shown in the picture, the two opposing E.M.F.'s would both force electricity out into the wire at the top of the armature, and draw it in at the bottom as shown by the arrows on these wires. This will cause a current to flow in the external circuit.

If the junctions of the coils are connected to eight commutator bars, (one bar per coil), and connect the ends of the external circuit by brushes to the commutator bars which are midway between the N- and S-poles, then each half of the armature separately generates an E.M.F. and delivers current to the external circuit.

Suppose the armature to be revolving at the highest safe speed. Each inductor will move past the magnet poles at a speed of 3,000 ft. a minute. With poles pieces 5×8 inches and a *flux density* of 90,000 lines per square inch, the *total flux* will be $5 \times 8 \times 90,000$ or 3.6 million lines.

The armature may be 9 inches in diameter which gives its rotative speed 1,270 (nearly).

For R. P. M.*=P.S.†÷(3.1416×diameter).

$$= \frac{3000 \times 12}{3.1416 \times 9} = 1270 \text{ nearly.}$$

which R.P.S.‡=21 nearly.

An inductor therefore cuts 3.6 million lines of magnetism twenty-one times a second, which is equivalent to cutting 75.6 millions once per second.

Since the cutting of 100 million lines per second by an inductor induces 1 volt pressure, each inductor on this armature revolving in this field will produce 75.6÷100 or $\frac{3}{4}$ of a volt (approx.).

The 4 coils of 4 inductors each (Fig. 233) on the N-half of armature being in series produces 3 volts per coil or a total of 12 volts which is the E.M.F. of the generator.

The S-half of the armature also generates a pressure of 12 volts, which is not added to the pressure of the N-half, being in parallel with it. An inspection of Fig. 233 shows that they oppose rather than add to each other; but an outlet being provided they turn aside through it, and send current separately and independently towards the outside circuit.

If the armature is wound with No. 10 A.W.G.§ the diameter of which is 0.102 inch or 102 mils, its area is 102 squared equal to 10,404 c. m. Allowing 700 c. m. per

*Revolutions per minute.

†Peripheral speed.

‡Revolutions per second.

§American Wire Gauge.

ampere, it will carry 15 amperes, without too much heating.

Since each side of the armature delivers its own current to the brushes, the safe current output of this generator is 30 amperes.

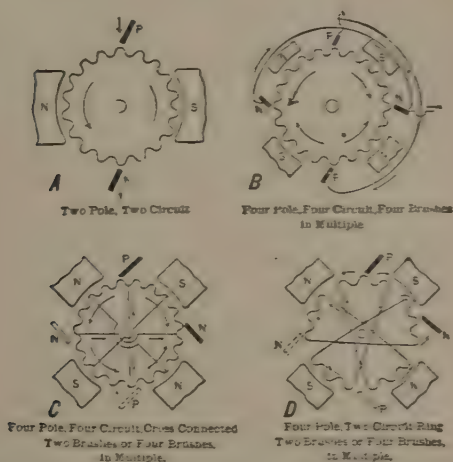


Fig. 234. Showing the Number and Position of Brushes on Different Armature Windings.

The black brushes are the ones actually used, the dotted ones being dispensed with on account of the particular winding.

Suppose there are 250 ft. of this No. 10 wire on this armature. The resistance of the wire from the wire table is 1.02 ohms per 1,000 ft.

The resistance of *all the wire* on armature is 0.255 ohm, and the resistance of the wire on each *half* of the armature is 0.128 ohm.

But the two halves are in parallel so *the resistance of*

the armature as measured from brush to brush will be half of 0.128 or 0.064 ohms.

The drop or loss of pressure in armature will be $C \times R$ or 30×0.064 , equal to 1.92 or say 2 volts.

This machine being a *shunt generator*, the main current does *not* pass through the fields, and there is no further voltage loss.

The E.M.F. of this dynamo is 12 volts and its *voltage* 10 volts. Its output in watts will be $10 \times 30 = 300$ watts or 0.3 kw. This is the *rating* of the generator.

The generator will carry this load 22 hours a day without getting more than 90° Fahr. hotter than the surrounding air.

A properly proportioned machine will stand a 25% overload for half an hour, rising an extra 30° in temperature, and it will stand a 50% overload for one minute without being damaged by the heat.

DRUM WINDINGS.

The extra labor involved in passing the dead wire through the bore of a ring core is avoided by going back to first principles again and placing on the core (either drum or ring) a number of coils shaped as in Fig. 227, producing a winding as in Fig. 229.

It is to be noted that the inductors lie entirely on the outer surface of the core and that the percentage of dead wire is less than in Fig. 231.

For a long, small diameter armature, drum winding uses the least wire; while for a short, large diameter core, the ring winding will require fewer pounds of copper.

In order to make the diagram in Fig. 230 clear, it has

its proportions wrong. The dead part of the wire is drawn very long and the active part very short. The reverse is true of an actual winding.

Referring to Fig. 229, and using Fig. 230 as a guide, the left side of the armature is the N-pole side and the right the S-pole side; and the armature is revolving anti-clockwise (otherwise the upper brush would be positive).

The E.M.F.'s on the N-side and S-side of coil T, as in Fig. 228, are in series and add up, producing a current flow towards the lower (positive) brush. The current passes through the inactive (dead) coil R in order to get to the positive brush.

At the same time the E.M.F.'s in coil B add up and passing through the dead coil L, drive current out of lower brush.

The value of the E.M.F. is eight times that which one inductor can produce. For the active coil T has 4 loops, i. e., 8 inductors in series, as also has the coil B. Suppose T produces 8 volts, the two coils T and B are in parallel and do not add their E.M.F.'s.

The coils L and R are dead, L being in series with B and R in series with T, but they produce no E.M.F. At the present instant they are but a wasteful resistance; their value, however, will be soon seen.

When the armature has moved about $\frac{1}{8}$ of a revolution, T is cutting flux slantingly and R, which is in series with it, is beginning to cut flux also. T is only $\frac{3}{4}$ active, producing say 6 volts, and R is not totally dead but $\frac{1}{4}$ active, producing 2 volts. Hence the voltage of the machine is still 8.

At $\frac{1}{4}$ revolution R is doing full work and B is dead and in series with it, while T is dead and L in series with it is at full activity. Now R and L produce the E.M.F.

The current enters the armature through the upper brush, splits and passes through the armature by two parallel circuits, one containing T and R in series and the other containing L and B. During a revolution these coils interchange places, but two coils are always in each circuit.

When 6 amperes flow in the external circuit the No. 16 wire of the armature is not overheated, as it has but 3 amperes to carry. It has 2583 C. M., which is more than 3×700 C. M.

Self-excitation of a Dynamo. When a dynamo is standing idle the field magnets are weakly magnetic, due to residual magnetism.

Let the armature revolve, and in a shunt, or compound machine open, in a series generator close the external circuit.

A few volts will be generated and cause a current to flow through the fields, hence the magnetism will increase and more voltage will be induced. This voltage will send increased current through the shunt field, and cause more volts to be induced.

The machine is now "building up."

As more and more magnetism is put into the fields, it becomes harder to get any more in as the iron is approaching *saturation* and there is more and more *leakage*.

Hence at a certain point, depending on the design of the machine, the difficulty of increasing the magnetism being added to the effect of the leakage just balances the tendency of the voltage to be increased. If nothing else is done the voltage of the dynamo will remain constant.

In the series field, is passing all the current drawn from the machine, and the field strength and voltage tend to increase. This increase is opposed by the C. R. loss

in armature and field, and the effect of the increasing field density. The net result is a *building up* of the voltage and if the load is not changed the voltage of the machine will remain constant.

Regulation. If now in the shunt generator the external circuit is closed, an extra current (very large in proportion to the field current) is drawn from the armature and causes a C.R. loss.

A lower voltage is thus impressed on the external circuit, also on the field. Hence the field weakens, and the added results of C.R. loss and weaker field is a considerable drop in voltage for each increase in load.

Resistance must be cut out of the field as load increases.

When in the series generator the load increases, a shunt should be placed around the field to weaken it, if a constant potential is desired.

Position of the Brushes. In order that one set of brushes may take away from, and the other set deliver current to the generator in a bipolar machine these sets are on opposite sides of the commutator.

In some dynamos when the inductors come out of the slots, one goes straight on to a commutator bar, and the other is bent over to its proper bar. This puts the brushes in line with part of the coil, and they will be found half way between the pole tips.

It is usual to bend both inductors as they leave the slots and connect to bars half way between the slots. Then the brushes will be found opposite the middle of the pole piece.

In dynamos and non-reversing motors the brushes are a little distance away from the points mentioned. but in reversing motors are exactly at these points.

The alternate brushes are of the same polarity and there is usually a set of brushes for each field magnet.

The placing of the brushes on the commutator with a certain relation to the winding is necessary as a reference to Fig. 234. or to the diagram of any winding will show that the brush while collecting current is at the same time *short circuiting* one of the coils.

In order that an excessive current may not be generated in this short circuited coil it must be out in the interpolar space at the time the brush touches the two bars belonging to it.

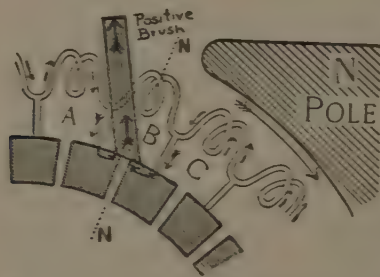


Fig. 235. Showing Position of Brush for Sparkless. Collection of Current.

Sparking. When a current is broken there is always a spark, which is greater the more turns in the wire, and the more iron within these turns. That is, the more *inductive* the current the worse the spark.

The conditions are right for excessive sparking in a machine, for the circuit is *inductive* and, although the circuit is not actually broken, the current being merely shifted, yet the result is equivalent to it.

Looking at Fig. 235 and considering the line N N to

be about midway between the pole pieces. The coil B is short circuited but has no current in it because:

1. The field is very weak and the coil is moving parallel to it, so no E.M.F. is generated in the coil.

2. The currents from the N- and S-side of winding enter the brush without going through the coil B.

Coil B has therefore no current in it, but being connected to A and C whose potential is high B is *charged* with electricity, and it is full of *coulombs*,* which are at rest.

When the armature revolves as shown and the toe of a upper brush leaves bar B the current from C must instantly change over going through B to reach the brush. The coulombs in B which are at rest should instantly move at full speed becoming a part of the armature current.

It being impossible to set the coulombs in B into motion instantaneously it is evident that the current from C encounters more than the *ohmic resistance* of the coil B. This extra opposition being called *reactance*.

The path through B being momentarily practically nonconducting, the circuit is broken by the brush moving away from the bar, and a spark or arc formed.

The circuit being *inductive* (having turns containing iron) the spark is persistent and holds until the *reactance* of coil B decreasing, it begins to conduct and diverts enough current into the proper path and the arc goes out for lack of current to maintain it.

This sparking is avoided in the following way:

1. Carbon brushes of high resistance are used which,

*A coulomb is a certain quantity of electricity. When a coulomb passes a given point every second a current of one ampere is said to flow.

as the part of the brush touching a bar gets narrower, due to the high resistance, throttles the current, gradually forcing it over to the coil B. Hence B does not have to instantly carry *all* the current.

2. Move the brushes of a dynamo in direction of rotation until they are nearer the pole shoe, exactly as is shown in Fig. 235.

The short circuited coil B is now under the *fringe* from the pole piece; and is moving obliquely through a stronger field. A small E.M.F. is generated in it.

From the illustration it will be seen that a current in the same, as in C (for B and C are under influence of same pole piece) flows around through B, the bars 2 and 3 and the brush.

By *shifting* the brushes a little to and fro the correct strength of field can be selected, and the obliquity at which it is cut adjusted, so that a current will be made to flow in B not only of *the same direction as that in C*, but also of exactly the *same value*.

Hence when the toe of the brush slips from bar 3 the current in C instead of running against the *impedance* (the sum of the resistance and reactance) of coil B, finds itself merely falling in behind the flow already established, and there is no tendency to spark.

In a motor the brushes are shifted in opposite direction to the rotation to get to the no sparking position. Hence the positions for sparkless forward or backward running are some distance apart.

It is a mere matter of first cost to produce a machine with absolutely *sparkless commutation* under any conditions. It is the skill of the designers which has (without prohibitive cost) so reduced the distance between these two points that it may be spanned by a thick carbon brush.

The railroad motor of today operates in either direction, without shift of brushes, under all loads, and some overloads, without serious sparking. What little occurs is of such small volume and such low temperature that no great harm is done.

CLASSES OF DYNAMOS.

Dynamos are divided into classes with reference to the manner in which their fields and armature are inter-connected.

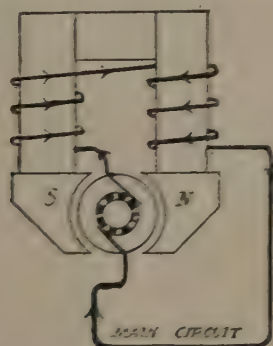


Fig. 236. Circuits in a Series Dynamo or Motor.

The series dynamo. Fig. 236. The same current traverses the field, armature and main or external circuits. The conductors in these circuits are about the same size. The circuits are all in series.

This dynamo is used for arc lighting and as a booster for increasing the pressure on a feeder carrying current furnished by some other generator.

The characteristic of this type is to furnish power at an increased voltage as the load increases. If sufficient

current is drawn to overload the machine the voltage will fall.

The shunt dynamo. Fig. 237. Here the field circuit is arranged as a shunt circuit. The armature and external circuits are in series. The armature current is the sum of the external and field currents. The conductors on the field are very much smaller than those on armature, as they only carry 2 to 5 per cent as much current.

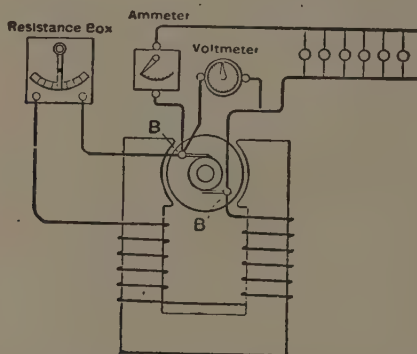


Fig. 237. Circuits of a Shunt Dynamo with Instruments and a Load of Lamps.

Used for incandescent lamp lighting, mill and factory power.

The characteristic of the shunt generator is to allow the voltage to fall as the load is increased.

It is evident that only by a combination of these two into a *compound dynamo*, Fig. 238, can a generator be produced which will deliver any power within its rated capacity and yet hold a steady voltage.

The armature is the same as a shunt dynamo, but

the fields have two distinct windings, one shunt and the other series.

The series dynamo is often called a *constant current* generator because its tendency is that way, and with a regulator it will furnish a constant current.

The shunt dynamo is similarly termed a *constant potential* generator. For with a regulator it will keep to a constant voltage.

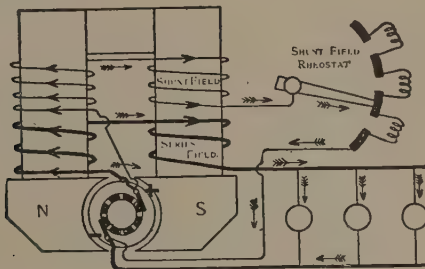


Fig. 238. Circuits in a Compound Dynamo.

The compound generator will of its own accord, without any regulator, furnish at its terminals, or at any distant point on the line steady power, at an absolutely constant voltage.

In railway service where the amount of power required fluctuates violently, the voltage will vary somewhat, for the generator scarcely has time to adjust itself to the present conditions before the condition no longer exists and a new demand arises.

It is not necessary that the pressure on the third-rail should be absolutely constant, and for this kind of service the compound generator, with the series characteristic predominating, so as to keep a steady pressure out on the line is best adapted.

PARTS OF DYNAMOS AND MOTORS.

The following is a short description of the various parts of a *dynamo* or *motor* and an explanation of the terms used in talking about them.

Base. Made of cast iron and supports the magnet yoke and bearings. It is made hollow to save unnecessary weight, and is braced internally with ribs to give strength. It is bolted to a masonry foundation.

Railway motors have no base; the yoke being bolted to the transoms of the truck, either directly or through a spring supported lug.



Fig. 239. Field Coil of Ribbon wound edgewise, with Laminated Magnet Core (on left).

Yoke. The ring or shell steel casting to which the magnet cores are bolted. Sometimes the cores are placed in the mold and the yoke cast around them.

In *railway motors* the cores are so short that they are frequently cast solid with the shell.

Pole pieces or faces. Generally of cast iron bolted or keyed to the magnet cores. Being greater in area than cores, they serve to hold the field coils in place.

In *locomotives* they are sometimes made of sheet iron.

Magnet cores. Very seldom made of sheet iron, sometimes of wrought iron forgings, but generally are steel castings.

Fig. 239 shows a field coil and a magnet core of an alternator. The winding of copper ribbon wound edge-wise. The pole piece is almost same size as coil.

Fig. 240 shows a shunt and series coil (large one is shunt coil) wound and taped ready to be slipped on the magnet core.

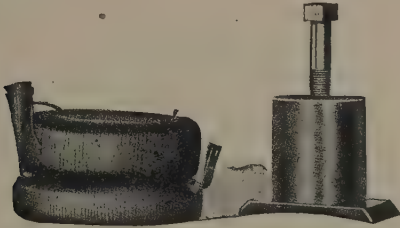


Fig. 240. Shunt and Series Coils ready to put on Core.

This core has the pole piece cast with a hole, which is bored to a driving fit on the core.

Fig. 241 shows a ring yoke with the field coils in place.

Fig. 242 shows one of these fields ready to be bolted in place.

Bearings. Should be large; four to six times as long as the diameter of the shaft they support, on account of the high speed. They are usually of the self-oiling type.

If there are collars on the shaft they should be arranged to allow the shaft to float, *i. e.*, to move to and fro about $1/16$ to $1/8$ of an inch so as to distribute the oil in the bearings, and to wear the commutator evenly. If there were no float, the brushes always touching on the same

line around the commutator, would wear grooves in the bars.

Rotary converters will not float of their own accord, so a magnetic or mechanical device is installed at one end of the shaft to compel a regular and even float.

At other end of shaft is often installed an "over speed" device to shut down power in case rotary "races."



Fig. 241. Ring Yoke with Field Coils in place.

Shafts are made thicker under the armature core than in the bearings; for the stress is greater out between the bearings, since the load acts with a lever arm.

They are made larger than in ordinary machinery because there is frequently a magnetic side pull due to

unequal strength of the field magnets, or the polar bore being eccentric with the shaft.

It is almost impossible to pull an armature out of the fields while current is passing through the field coils.

Core. The sheet iron body upon which the armature winding is placed.

Fig. 243 shows a toothed core into whose slots the armature winding is laid. The core is held firmly between two end plates, by bolts passing through tubes of insulation (usually fibre).

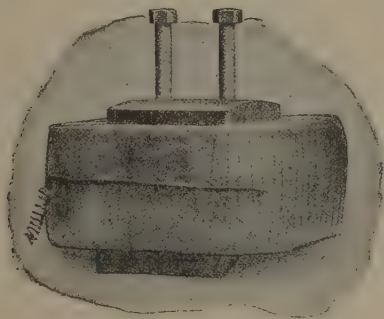


Fig. 242. Field Coil of Fig. 241 ready to be bolted into place.

Generally the armature core is held by a *spider*, whose hub is turned and keywayed to receive the commutator hub. The spider hub is also bored, reamed and keywayed so as to form a *sleeve* or *quill*, which is slipped on the main shaft. In this case the complete armature may be removed for repairs without disturbing the commutator leads or getting core and commutator out of alignment.

Fig. 243 shows this construction.

Hysteresis. When the iron core revolves it passes in succession under N- and S-poles and so the magnetism of the core is reversed.

All iron, no matter how soft, holds some *residual magnetism*. This "hanging on" of the magnetism is called *hysteresis*.



Fig. 243. Armature Spider and Toothed Core.

Before the magnetism in the core can be reversed, the residual magnetism must be wiped out. It takes about $1\frac{1}{2}$ per cent. of the total power of a generator to do this work. This inevitable loss is made as low as possible by the use of a small quantity of the softest iron, worked at a low magnetic density. Increasing the quantity of magnetism per square inch, the speed of the machine, or

the pounds of iron in the core *all* increase the loss due to hysteresis.

Eddy currents, (sometimes called Foucault currents.) There must be E.M.F. induced in the iron of the core, just the same as in the copper on the core. Were it not that the core is in sheets placed parallel to the flux, there would be a great waste of power sending current through a short thick body of iron. (Hence of low resistance.)

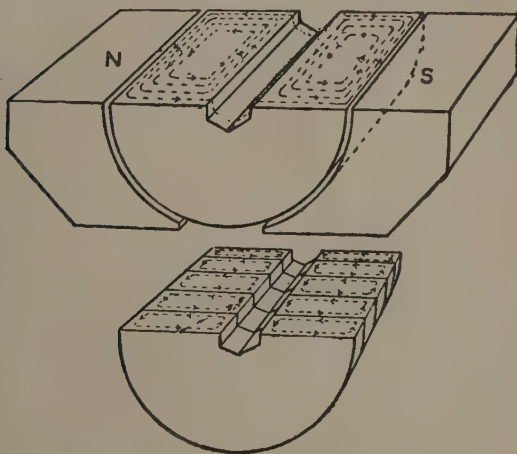


Fig. 244. Effect of Solid and Sheet Iron Construction on Eddy Currents.

The insulation of rust, etc., between the sheets forms a series of short, very thin, conductors, which are of high resistance, and hence the eddy currents are small. An increase of speed makes a great increase in the eddy currents. The usual loss due to them is about $1\frac{1}{2}$ per cent of the total power generated.

Fig. 244 shows how the paths for eddy currents are made long and narrow by using sheet iron.

ARMATURE WINDINGS.

Ring. These windings are very simple to calculate but hard to place on the core. The method of winding is shown in Fig. 245.

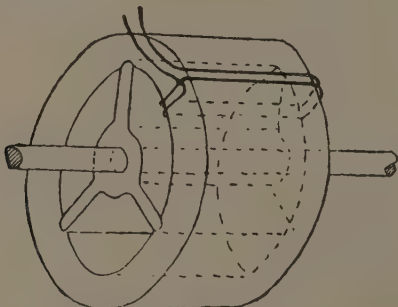


Fig. 245. Ring Winding.

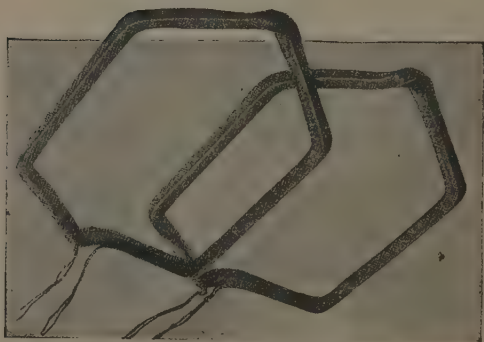


Fig. 246. Formed Armature Coll.

Drum. The double cotton covered wires are bent into shape, insulated with linen tape, baked and shellac varnished. A number of these *formed coils* (Fig. 246) are



Fig. 247. Toothed Core with Formed Coil Winding.

placed in the slots (Fig. 243) on the outer surface of the core (Fig. 247, also Fig. 248) and their ends soldered to the commutator bar lugs (Fig. 249).

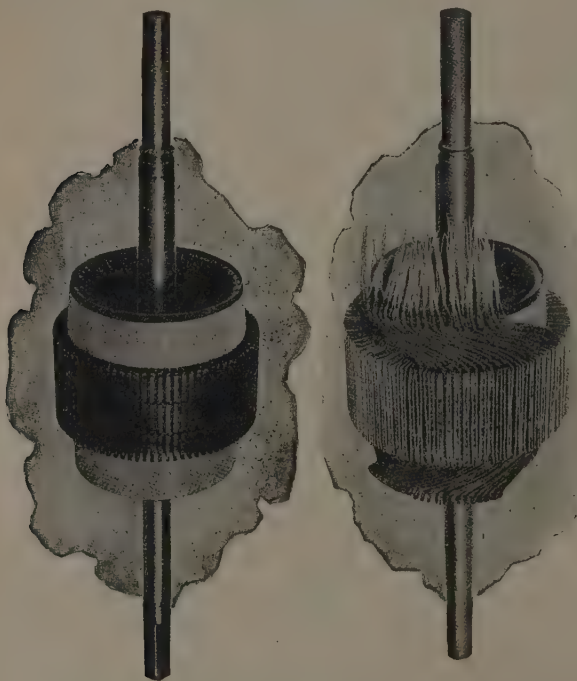


Fig. 248. Formed Coil Winding Incomplete, and Completed.

Each slot contains two inductors. One side of a coil is in the bottom of a slot under a N-pole, and the other side in the top of a slot under a S-pole. This same slot has in its bottom a second coil, the outer side of which is in the top of a slot under a N-pole. The first mentioned coil has its two ends connected to any two adjacent com-

mutator bars, and the second coil is connected in the same manner to two bars on opposite side of commutator.

Commutators. The peculiar shape of the bar (Fig. 250) makes it possible by the corresponding shape of the hub and washer to draw bar and insulation (mica) together tightly, both sidewise, endwise, and downwards, locking them absolutely in place by a nut bearing against the washer.

The hub is sometimes keyed to the shaft. This gives a chance to remove the commutator for repairs, by unsoldering the leads from armature winding.

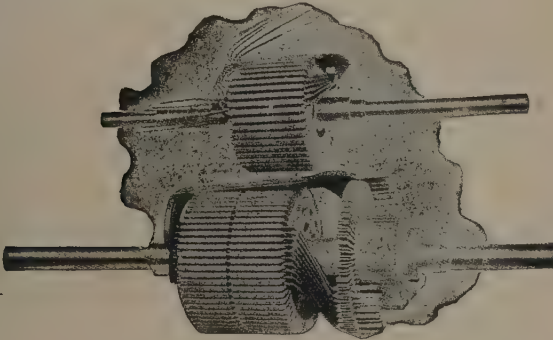


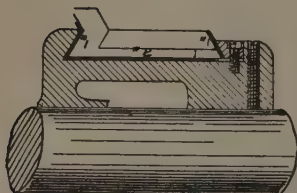
Fig. 249. Armature Winding connected to Commutator.

A complete armature, i. e., wound core and commutator, is shown in Fig. 251 on a quill ready to be put on axle or shaft.

The golden copper color of a new commutator soon changes to a uniformly rich dark mahogany brown, and its surface acquires a polish. The degree of the polish and uniformity of the color are indications of the success of operation. Any blackening of the bars or rough-

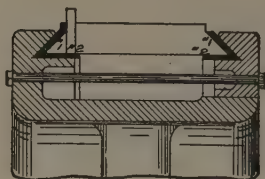
SIEMENS-HALSKE.

TAPER RINGS #1
SLEEVE FOR SHELL #2



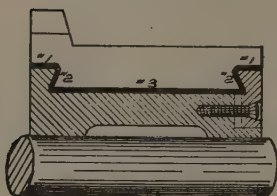
FT WAYNE.

TAPER RINGS #1
BAND RINGS #2



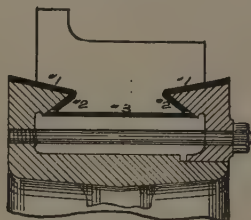
EDISON BI-POLAR

BAND RINGS #1
TAPER RINGS #2
SLEEVE FOR SHELL #3



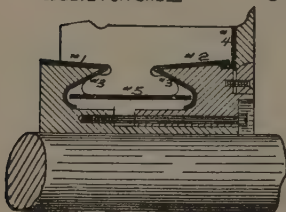
WALKER.

TAPER BAND RINGS #1
TAPER RINGS #2
SLEEVE FOR SHELL #3



G E. 800

TAPER BAND RING NARROW #1
TAPER BAND RING WIDE #2
TAPER RINGS #3
FLANGED FLAT RING #4
SLEEVE FOR SHELL #5



WESTINGHOUSE

TAPER BAND RINGS #1
TAPER RINGS #2
FLAT RING #3
SLEEVE FOR SHELL #4

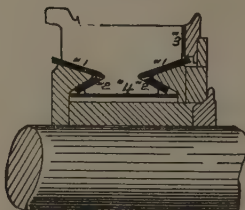


Fig. 250. Construction of Commutators Showing Hub, Mica and Bar.

ness of surface shows defects in machine or manner of operating.

Brushes of copper or brass in the form of sheets or folded gauze were formerly much used. In the modern machines with the losses reduced by clever designing, more current is drawn, a higher voltage generated, and larger pole pieces used on the same sized armature.

The tendency of the brushes to spark is thereby increased, and to offset this *high resistance* carbon brushes must be used. Hence the use of carbon brushes is universal. Where the tendency to spark is not so great graphite brushes are used.

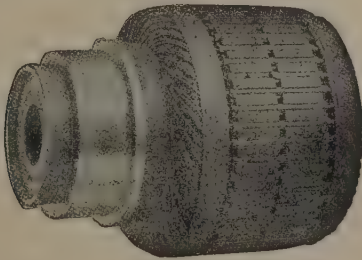


Fig. 251. Complete Armature.

For mechanical reasons copper rubbing on copper wears worse than carbon on copper. The abrasion between two surfaces of same material is generally worse than two different ones.

Dynamo brushes sometimes set at a slight angle from being perpendicular to the commutator, as they always revolve in the same direction.

For a motor the brushes are set radially against the commutator, for otherwise it could not be reversed with-

out danger of breaking the brushes. They are pressed against the bars at about $1\frac{1}{4}$ lb. per square inch.

Sufficient area of carbon must be actually touching the commutator to collect the current at a density of 30 to 40 amperes per square inch.

This total area is divided up into small blocks about $1\frac{1}{2} \times 2$ or 3 inches, and each has its own spring. This insures that at all times all of the brush surface is in contact with the commutator.



Fig. 252. Brush Holder.

Brush holders. The holders must clamp the brushes securely, and press them straight against the commutator, as in Fig. 252. There must be sufficient area of contact between holder and brush, with good pressure so that the current may pass from brush to holder without producing a great C. R. loss.

The springs maintaining the pressure of the brush against the bars should not carry current, as they might heat and have their elasticity destroyed.

The current carrying capacity of the brush holders is often supplemented by *stranded* or *braided* conductors of copper, running from the brushes back to the brush holder cables. These are called "pigtails."

A set of brush holders is mounted on a brush holder arm or stud.

The brush holders on opposite sides of the commutator are not in the same plane, but by having one stud a little shorter than the other the brushes are *staggered*.

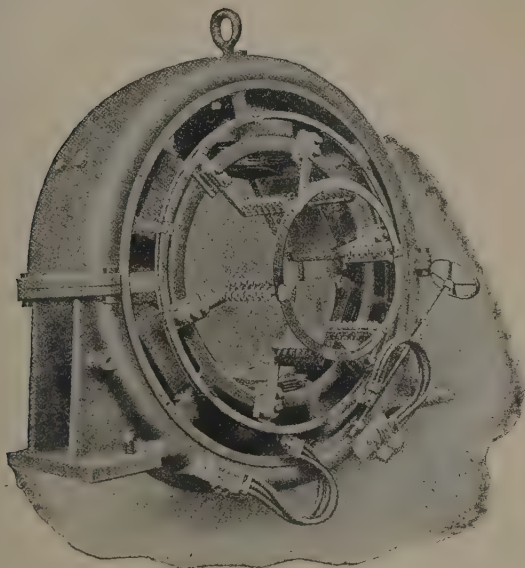


Fig. 253. Complete Brush Rigging on a D. C. Generator.

There is then no part of the commutator not worn by the brushes, and this together with the *float* of the shaft, keeps the commutator a perfectly smooth cylinder of slowly decreasing diameter as it wears.

An oval, barrel, or spool-shaped commutator due to improper wear must be swung in a lathe and light cuts

taken until it is again of the correct shape. This is called *turning down*.

The brush holder studs project from a ring which slides in grooved arms supported by the bearing pedestal



Fig. 254. Brush Holder on R. R. Motor.

or by the field frame. They can thus be moved simultaneously and set into the position of *no*, or minimum sparking for full load, then locked in place. See Fig. 253.

On *railway motors* each holder stud is often clamped temporarily to the field frame, or transoms of truck until

the correct position is obtained, when they are bolted permanently directly against frame or transom.

The ability to change their position simultaneously is thus lost, but simplicity, and reduction of number of parts and weight is obtained.

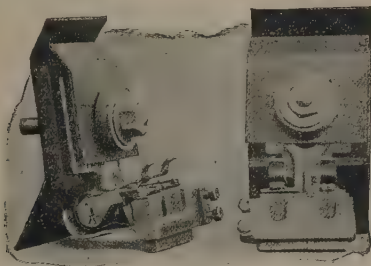


Fig. 255. Brush Holder on a R. R. Motor.

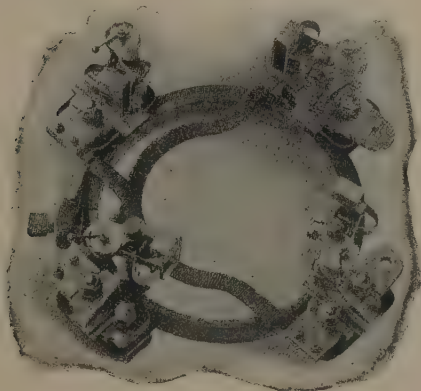


Fig. 256. Set of Brush Holders and Collecting Cables.

Figs. 254 and 255 show brush holders of railway motors.

In Fig. 256 is shown an arrangement similar to Fig.

253. It is for a smaller dynamo and has no hand wheel to set brushes, but merely a set screw clamp shown on left side.

Field coils. The spools in which these coils are wound are made of brass or bronze, which being non-magnetic does not cause any leakage through its flanges.

Copper *strip* or *ribbon* is used on railway motors and fuller board or card board insulation between turns. Fig. 239.

To prevent injury to the windings they are usually wound with a layer of cord or tape. Fig. 257.

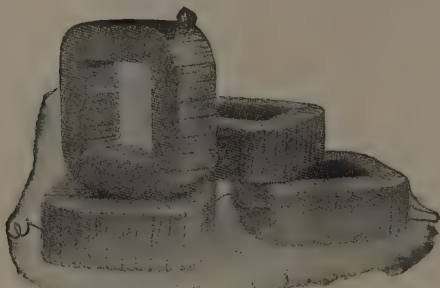


Fig. 257. Field Coils taped ready to place on Magnet Cores.

Fig. 258 is a railway motor field coil wound of strip copper and protected by a layer of cord. In motors where the field frame is not a shell, completely enclosing the field and armature, protection is furnished to the coils by brass shells. These are riveted to the flanges of the field spool.

In the New York Central locomotive two hand holes

are left, one containing the two terminals of the coil. Into these holes are poured a hot bituminous insulating compound, and when cool the covers are screwed on.

From transom to transom under the armature is hung a curved piece of brass to protect it from flying stones and pieces of iron. Being perforated with 5/16 holes it does not interfere with the passage of air to cool the armature.

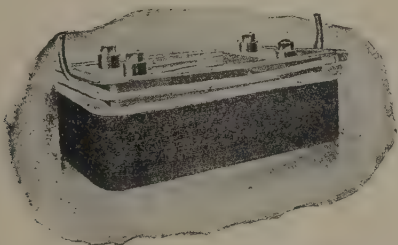


Fig. 258. Field Coil Wound with Strip Copper Insulated with Asbestos.

Rheostat. In a dynamo an extra resistance, the value of which is about 25 per cent. of that of the fields, is placed in series with them, and by cutting more or less of this *field rheostat* in or out of the circuit, the field current is strengthened and weakened, and the voltage of the generator increased, or diminished.

At least part of this resistance is always in circuit, and it is so designed as to size of wire and ventilation as to do this and not overheat.

Fig. 259 shows a rheostat for a railway generator. The handle projects through the switchboard, while the resistances made of cast iron are behind the panel. In Fig. 260 is shown the operation of such a rheostat. The

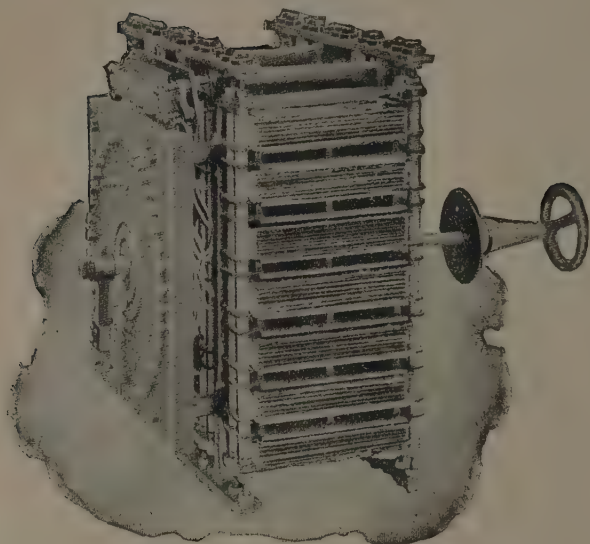


Fig. 259. Field Rheostat for Railway Generator.

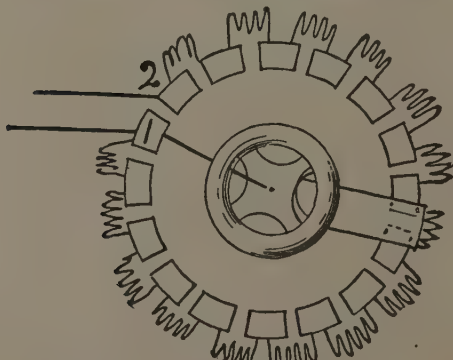


Fig. 260. Diagram of Field Rheostat.

current enters at 1 and goes through half of the resistance and out at 2. The amount of resistance in circuit is thus regulated by position of handle.

ALTERNATORS.

The main difference already noticed between the D. C. generator and the A. C. generator, called for shortness' sake an alternator, is that one has a commutator, and the other a collector.

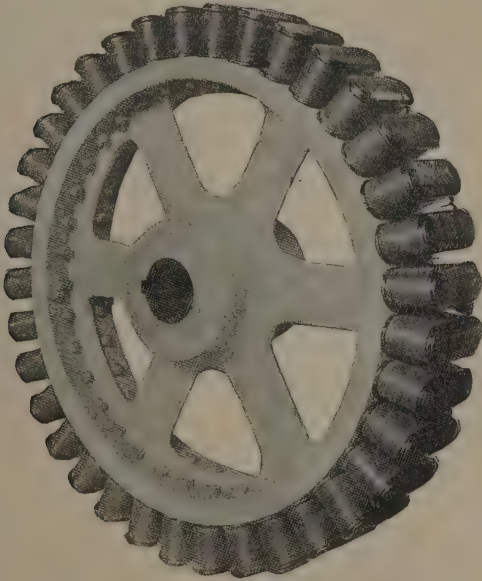


Fig. 261. Revolving Field of Alternator.

There are, however, differences in construction which must be noticed.

The highest voltage for which D. C. generators are

wound is 1,200, this being the lowest voltage for which alternators are wound, while for railroad work 11,000 is the usual, and 22,000 not uncommon.

This makes the problem of proper insulation for A. C. armatures more difficult. To make the work easier, in-



Fig. 262. Stationary Armature of Alternator.

stead of having the field stationary, and armature revolve as in most D. C. generators, in alternators the field revolves and the armature is stationary.

The field is fed with D. C. at 250 volts pressure and is easy to insulate, even though subjected to the mechanical strains of rapid motion and the lack of plenty of space.

The armature being stationary, there are no mechanical strains, also weight being no objection plenty of space can be given to insulation.

This type of construction is shown in Fig. 261 and 262. Fig. 261 shows a revolving field of 18 pairs of poles or 36 poles. Current is led in through a collector. This field revolves inside of the stationary armature (Fig. 262) whose windings are fully exposed to the cooling effect of

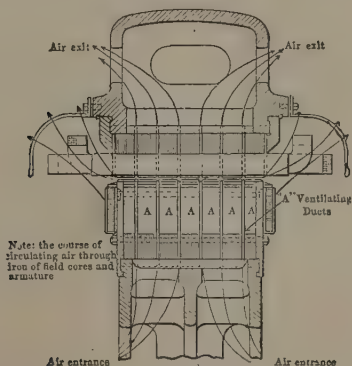


Fig. 263. Method of Ventilating Alternator with Revolving Field and Stationary Armature.

the air. This armature needs no collector, for the terminals of the winding are attached to leads which come out of the base at one side.

Ventilation of the field and armature is accomplished by means of air ducts, as is well shown in Fig. 263.

The magnetism in all armatures fluctuates and reverses in polarity as it passes, or is passed by the poles; but in alternators these reversals of magnetic polarity are much more rapid than in D. C. machines.

To prevent the waste of power necessary to magnetize and demagnetize solid iron at such a rapid rate, not only is the armature iron laminated but the magnet cores also.

Fig. 264 shows such a magnet core and Fig. 265 shows the way these cores are attached to the magnet frame or yoke.



Fig. 264. Pole Piece and Core of Laminated Construction.

THE EXCITER.

To furnish the field current for all the alternators of a station two small D. C. generators are used, one being run while the other is being cleaned, or held in reserve.

When storage batteries are in a station it is often the custom to make each exciter of such a size as to carry continuously three-quarters of the field load.

When a station is fully loaded, *i. e.*, all alternators running, both exciters are run, and when the load is light one exciter is shut down. In cases of exciter trouble the storage battery furnishes the field current.

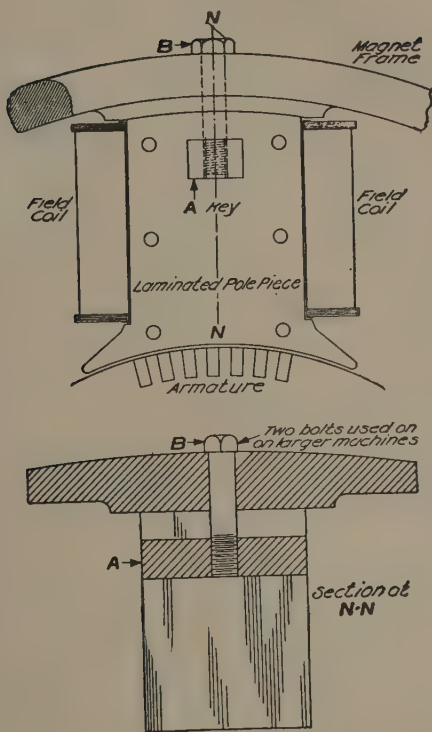


Fig. 265. Attaching Cores to Yoke.

SINGLE PHASE.

The field coils occupy about 50% of the surface of the field bore, because when their inner edges are tight together their outer edges are apart, due to the larger circumference at the pole pieces, and because some inter-

polar space must be left to prevent excessive leakage from pole to pole.

Only 50% of the armature bore is wound, for otherwise the coils would be so wide that they would extend over into the field of a wrong pole piece. If one side of a coil is under a N-pole the other side should be under a S-pole. Then the two E.M.F.'s induced, add together, Should the coil be so wide as to extend over to the next N-pole any E.M.F. induced by that pole would be subtracted.

There is then on the ordinary alternator half of the armature empty. Such a machine is called a Single Phase Alternator.

TWO AND THREE PHASE.

It occurred to some inventor that an entirely separate winding could be put on between the coils of the original winding and be connected to its own collector. The current was to be led to a different circuit, but it soon became evident that it was better to make of the four wires from the alternator, a three-wire circuit by joining two of them inside the armature and leading out three wires to the switchboard. Such an alternator is a Two Phase Alternator.

Of course the capacity of the machine is not doubled, because from a single phase alternator is drawn enough current to heat it to the safe limit. From a two phase alternator we do the same thing. The reason we gain in capacity is because in a single phase machine the heating is concentrated, while in the two phase machine it is evenly distributed all over the armature.

Even in a two phase alternator there is a portion of the armature not used for winding and there was still a desire to reduce the number of line wires. This led to the Three Phase Alternator.

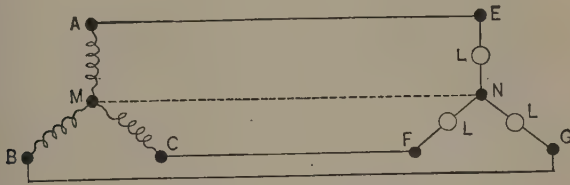


Fig. 266. 3 Phase Y Connection.

The three armature windings of the alternator are connected together at one point, and the other ends to the three collector rings, or the three windings are connected in series and the three points where they are joined are connected to the three collector rings.

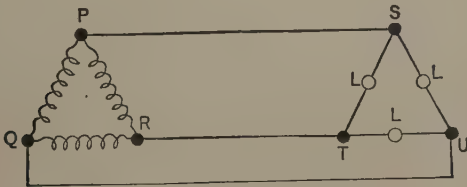


Fig. 267. 3 Phase Delta Connection.

The former winding is called a Y winding and is shown in Fig. 266. The latter is a Δ (Delta) winding and is shown in Fig. 267. The European names are respectively Star and Mesh windings.

The three wires of a three phase system each act as a main wire, and a return wire for one of the others at the

same time. The actual current in the wire is the difference of the two currents: in and outgoing.

If the same three phase armature is connected first as a Y and then as a Δ winding these differences will be noticed.

The Y armature will give the higher voltage and have less current capacity. The Δ will give a lower voltage and have greater current capacity. Power that can be drawn from each is the same.

Transformers and other apparatus are wound two, and three phase, and also Y and Δ , for use with the correspondingly wound alternator.

By a peculiar connection of coils rotary converters are wound for six phase currents; it having been discovered that it is possible to do so with the result of increased output for a given sized machine.

Two, three and six phase machinery is often grouped under name of polyphase.

RAILWAY GENERATORS.

Undoubtedly the feature of greatest importance in all railway generators is reliability of operation. On account of the enormous fluctuations in the load thrown upon a railway generating plant, the generators must be capable of carrying great overloads, at least for short periods, without injury either from heating, or from sparking. In order to give the student an idea of the construction and operation of the various types of modern railway generators, a short space will be devoted to descriptions and illustrations of a few of these machines.

The question of whether a direct-connected or belted generator shall be used depends upon the particular requirements of the plant for which it is intended. To meet these various demands the leading manufacturing concerns build generators adapted to the service, as for instance, Allis-Chalmers Company produce three separate types of machines, known as "H," "III," and "I." The two former are usually belted, although well adapted for direct connection to alternating-current motors supplied by high tension transmission systems; the last named type is built for mounting directly on the engine shaft. Following is a brief description, with illustrations of the most important features of these machines:

Frame. The field frame is a circular casting usually made of cast iron, of elliptical cross-section. This yoke is divided horizontally in a plane passing through the shaft. The two parts are held together by cap screws whose heads are set in pockets covered by the name plates.

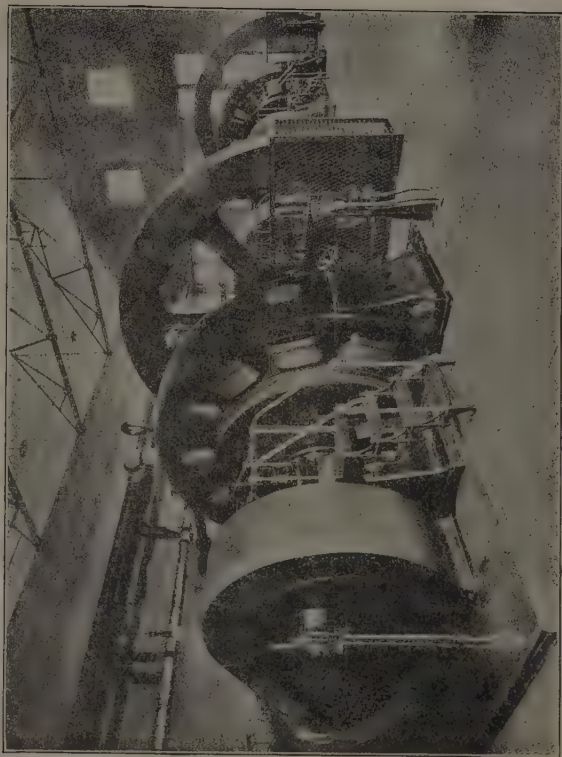


FIG. 268. TWO ALLIS-CHALMERS ENGINE TYPE RAILWAY GENERATORS.
Installed in the 32nd and Dauphin Sts. station of the Philadelphia Rapid Transit Co., Philadelphia, Pa.

The inside of the yoke is bored concentric with the armature shaft and to this finished surface the poles are bolted by cap screws, whose heads are countersunk flush with the external surface of the yoke.

Poles. As a rule the poles are built up of punched steel laminations, alternate laminations being reversed, so as to form a pole of the form shown in Fig. 269. This form of construction practically eliminates the eddy currents that would be produced in solid pole faces if wide open slots and short air-gaps were used. In a few units, it has been found expedient to use round cast steel poles,

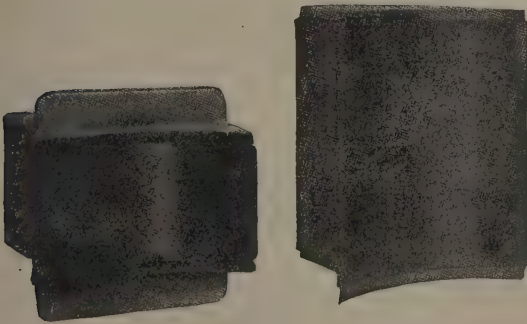


Fig. 269.

which have usually been provided with laminated steel pole-faces. The field coils, after being wound in forms which match the poles, are carefully and thoroughly insulated, the shunt and the series coils being made up separately. The shape of the poles is such as to hold the field coils firmly in place, the series coil being usually placed nearest the armature. As the poles are held in place by cap screws, it is possible to remove them to re-

pair either a field coil or an armature coil, without disturbing the rest of the machine.

Armature. The armature core is built up of punched steel laminations mounted on a cast metal spider. In the larger sizes, the rings are made up of segments which

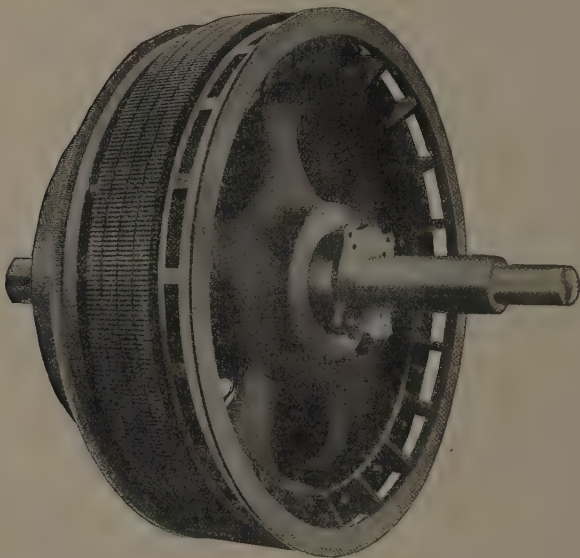


Fig. 270. Armature Core and Commutator with Temporary Shaft.

are built up on the spider in such a way as to obtain the greatest possible strength. The armature construction is very open and ample air passages are provided throughout the core to insure thorough ventilation. See Fig. 270. Thus the heat in the armature is dissipated as fast as generated, permitting heavy overloads to be carried

without dangerous heating. The armature coils of all railway generators are made of rolled copper strip bent into shape in such a way as to be interchangeable on the same armature. The several conductors that are to go into the same slot, after being insulated individually

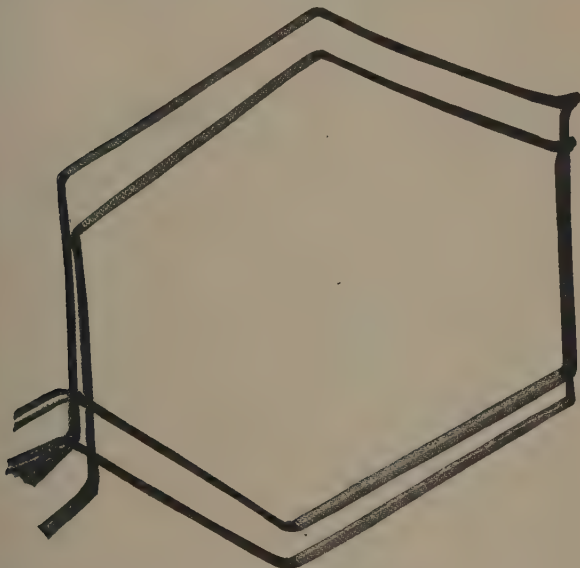


Fig. 271. Armature Coils for Multiple Winding, Types "H" and "HI" Generators.

with linen tape, are placed together and have the necessary insulation wrapped around them as a group. This insulation is pressed into shape in a steam heated coil press, after which it is cooled while in the press and the group of conductors is then ready to be placed in the armature slot. Figs. 271 and 272 show armature coils of

this kind ready for use. The armature coils are held in the slots by wooden wedges fitted into grooves near the top of the armature as shown in Fig. 273. At each end outside of the magnetic field, the armature coils are held down tightly by steel wire bands.



Fig. 272. Armature Coils for Series Winding of Types "H" and "HI" Generators.

Commutator. The commutator segments are made of hard drawn copper, tapered to the required dimensions in the drawing process. The segments are insulated from each other by mica insulation approximately .040" thick and are insulated from the clamping ring and sleeve by mica insulation .1" thick. Whenever it is practical to do so, the connecting pieces between the commutator segments and the armature conductors are made of thin

strips of rolled copper fastened to the segment. These open necks serve to blow air drawn from the inside of the armature over the face of the commutator and keep down its temperature.

Brushholders. The brushholders are of the reaction type having an adjustable spiral spring. The brush is held to the holder in such a manner that it may move

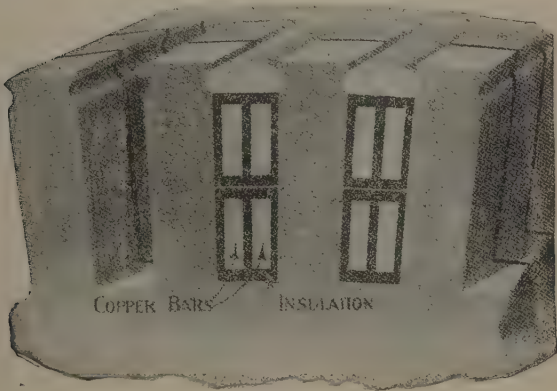


Fig. 273.

freely parallel to the face of the holder, but it cannot get away from the brushholder even though the tension finger is raised, until it has been lifted clear of the holder. With this form of brushholder, the brush follows most easily any irregularities there may be in the commutator, because the inertia of the moving parts is reduced to a minimum; only the brush and the tension finger move; the brushholder itself is stationary.

Brush oscillator. The effect of the lengthwise motion of belted armatures in producing highly-polished com-

mutators is well known, and manufacturers of belted machines take every precaution to secure this advantage. With generators direct connected to the engine shaft, however, the case is different, and as it is impossible in

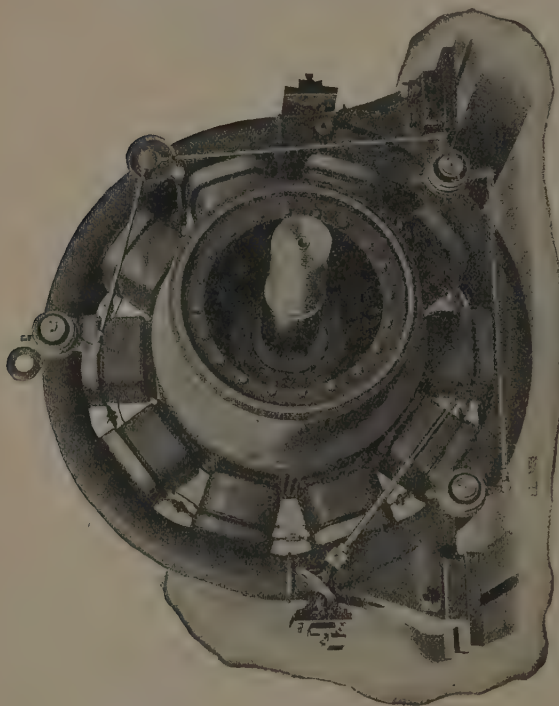


Fig. 274. Allis-Chalmers Engine Type Railway Generator with Brush Rigging off, Showing Patented Oscillating Mechanism.

such machines to move the armature lengthwise, it becomes necessary to move the brushes and brushholders back and forth in order to keep an evenly polished commutator surface. On the Allis-Chalmers direct con-

nected machines the brush oscillator is constructed as follows, referring to Fig. 274: The yoke which carries the brushes and brushholders is an iron casting of circular form supported by three or more cast-iron rollers fastened to the field frame. These rollers are moved back and forth approximately $\frac{3}{8}$ " by means of internal cams, which are driven through suitable gearing by a belt running over the engine shaft at the back end of the armature. By the use of this ingenious mechanism the brushes are moved slowly across the face of the commutator, effectually preventing the cutting of grooves. This device is provided with all engine type generators having armatures forty-five inches in diameter or larger.

COMMUTATING-POLE GENERATORS, BOSTON ELEVATED RAILWAY.

Among the comparatively recent improvements made in the design of direct-current machines, probably none has done more to mark a distinct advance in the art than the employment of commutating poles. While commutating-pole machines of moderate capacity have been in successful operation for some time past, it is only recently that this feature has been applied to large direct-connected engine-driven generators operating under the severe service conditions generally met with in street railway work. Fig. 275 shows one of two 2,700-kw. commutating-pole railway generators recently installed in the Charlestown and Harvard Power Stations of the Boston Elevated Railway Company by the General Electric Company and presents some interesting features in connection with the design, installation and operation of this type of railway generator. The following description,



Fig. 275. General Electric Commutating Pole Generator in the Power Station of the Boston Elevated Railway Co.

together with the illustration, are taken from the Street Railway Review of February, 1908: Both machines are direct connected to McIntosh & Seymour 4,100-hp. vertical cross-compound engines. They are designed for 575 volts at a speed of 90 revolutions per minute, but are capable of wide voltage range at all loads, and at the voltage for which they are designed are guaranteed to deliver 2,700 kw. continuously with a temperature rise not to exceed 35 degrees C. in any part above the surrounding air, also 50 per cent overload or 4,050 kw. for two hours following the normal run, the temperature rise not to exceed 5 degrees C., and to withstand a momentary overload of 100 per cent or 5,400 kw. without injury.

No change whatever in the brush setting has to be made from no load to momentary overload condition of 100 per cent, and the circuit breaker can be tripped under any condition of load between these limits without appreciable disturbance at the brushes. In fact, the provision for shifting brushes is almost superfluous on this type of machine, as with the brushes once adjusted for best commutation it is never necessary to alter their position, sparkless commutation being assured under all conditions of load. Moreover, the machines may be successfully run in parallel with other machines of the non-commutating pole type, no trouble whatever being experienced in paralleling the machines with others in the Boston stations.

The magnet frame is of cast iron, rectangular in shape, made in six sections securely bolted together and of the box section type, forming a most rigid structure for supporting the numerous poles with their heavy field coils. The main poles are rectangular cast steel, with laminated pole-faces, and the commutating poles are laminated

throughout. Field coils are compound-wound and of the well-known G. E. ventilated type.

The armature core, formed of steel punchings, is rigidly held to a cast-iron spider of substantial proportions by means of numerous dovetails. Effective cooling of the armature is procured by numerous ventilating ducts throughout the core, and also by a novel construction of the end flanges which allows the end conductors to be well separated, and permits a free and natural flow of air between them. The armature winding is of the multiple drum type, composed of formed coils of bar copper, and interchangeable on the same armature. The coils are held in the slots by tough wooden wedges and the end-windings are kept in position by patented sectional binding bands, easily removed or replaced.

The commutator bars are of hard-drawn copper of such dimensions as to provide ample wearing depth and liberal creepage distances. The spider, made of cast iron, is rigidly bolted to the armature spider and the clamping rings are of the best cast steel. An interesting feature of the commutator construction is the precaution taken to prevent a possible short circuit from studs or bars to clamping rings. Where the clamping rings extend beyond the commutator bars, they are thoroughly insulated with mica and in addition, extra insulation is molded over the extreme edge of ring and carried well down underneath, where it is securely fastened, thus making it almost impossible for grounding at this point.

The brush rigging is of the latest type, employing cast brackets of substantial dimensions to which the individual brushholders are bolted. The resulting construction is strong, rigid and entirely free from movement or vibration, thus obviating the necessity of employing support-

ing tie-rings passing from stud to stud. Alternate sets of brushes are so placed on the stud brackets that they are staggered around the commutator. By this means the brushes do not track, thus promoting even and uniform wear of commutator without the use of auxiliary means to obtain this desirable feature.

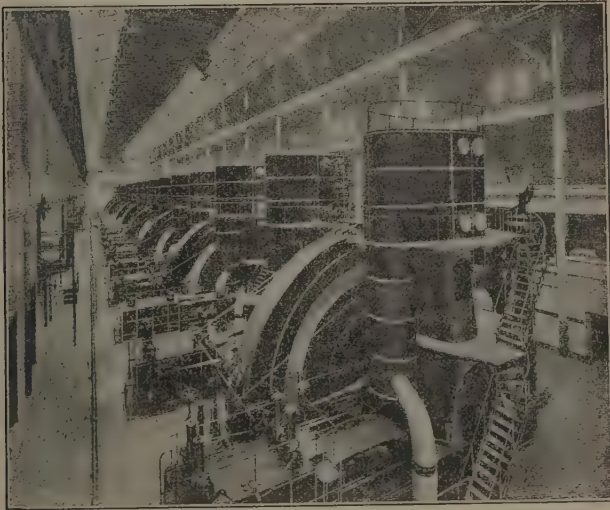


Fig. 276. Set of 5,000 K. W. Engine Alternators, Interborough Plant, New York.—Westinghouse Revolving Field Alternators.

REVOLVING FIELD ALTERNATORS.

The use of alternating current apparatus is rapidly increasing. As a means of transmitting energy over long distances and distributing current through extensive districts, the system stands alone. As an economical, con-

venient and efficient method of distributing power in manufacturing and industrial operations, it possesses many elements of superiority and is rapidly growing in favor. The economies permitted by the use of higher voltages, the ease with which these voltages can be transformed, and the simplicity, strength and endurance of the apparatus, are features of the system which commend its general adoption.

Fig. 277 shows the field core of a Westinghouse revolving field alternator. Current is generated in coils wound upon a stationary armature and connected directly to the supply circuit, without the interposition of moving contacts of any kind. The absence of collector rings and brushes in the main circuits is particularly advantageous in machines of large current capacity or high potential. Insulation can be more readily applied to coils which are stationary and therefore free from centrifugal strains, and the revolving field construction permits more room for winding and insulation and produces a machine of greater mechanical strength.

The method of construction of the rotating field is similar to that of the armature, except that the laminæ are dove-tailed into a rotating spider instead of a stationary frame (see Fig. 277). Ventilating ducts are provided in the core of the field which register approximately with the corresponding ducts of the armature.

The field coils are wound to slip over the projecting poles, and are held in position by wedges, which are driven between the poles in the slots near the tips. These wedges are made of copper and, besides holding the coils, serve as dampers to reduce any "hunting" action which may be set up between generators running in parallel, or between a generator and synchronous motors, or rotary

converters operating from its lines. These dampers are sometimes made in the form of a grid. The field coils are wound in molds. Copper strap bent on edge is used whenever possible. Square wire is used for the smaller sized generators, and for machines having bolted on poles. The strap winding is insulated between turns with asbestos strip. The wire coils are wound dry, and treated with insulating varnishes.

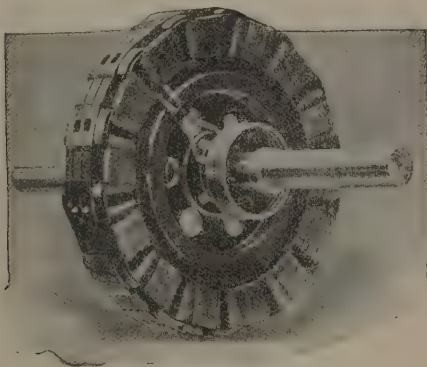


Fig. 277.

Fig. 262, on page 568, shows a view of a stationary armature. The core is formed by assembling dove-tailed punchings of soft sheet steel within the slotted frame. At suitable intervals—depending upon the capacity and size of the generator—the punchings are separated by ventilating plates which form ducts in the core. The punchings are slotted to receive the winding. The winding of the armatures is of the open type. The coils are formed in moulds on the machine, or are separately wound, shoved through the slots and afterward connected

—depending upon the voltage and current capacity of the generator. The separate circuits of the two-phase machines are normally insulated, but may be connected within the machine. When so inter-connected, the voltage across the corners is .7 of that across each phase.

Terminals. Flexible leads are brought through the frame near the bottom and are connected directly to the line.

Excitation. Westinghouse alternators are usually separately excited at 125 volts; 250 volts may be used if occasion requires. The lower potential is to be preferred because it lends itself more generally to the use of the strap wound field coil and is more convenient for the operation of lights and auxiliary apparatus in a station. The exciters are standard d.c. generators either belt driven or direct connected to engine, motor, or water wheel.

Composite Winding. Small high speed A.C. generators are sometimes composite wound, the compensation being obtained by means of series transformers connected in the armature circuit and having their secondaries connected to the separate field winding through a rectifying commutator which is mounted on the generator shaft.

Collectors. The collector rings of standard alternators are built up on separate spiders which are bolted to the field hub. They are of the skeleton or open type to secure a maximum ventilation. The rings are usually made of cast iron. The brushes are of carbon. These collectors carry only the low voltage direct current which excites the field. Fig. 278 shows a spider type collector.

Commutator. Composite wound generators are provided with two-part commutators mounted near the collector rings.

Ventilation. The losses which of necessity occur in any electrical machine appear as heat, and produce a rise in temperature which continues until equalled by the rate of radiation. The rate of radiation depends upon the difference in temperature between the generator and the surrounding air, and on its ventilation. It follows, therefore, that, with a given loss, good ventilation will decrease the rise in temperature.

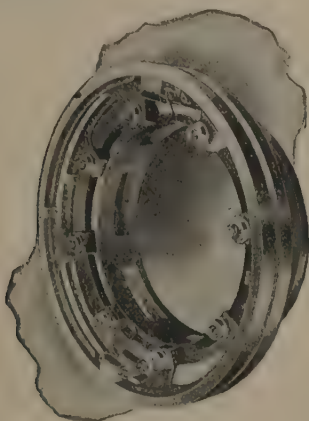


Fig. 278.

In the Westinghouse generators the air is entrapped by the ducts in the revolving field and forced outward by a centrifugal fan action through the corresponding ducts in the armature core and escapes through the openings in the cast iron frame. Ventilators at the end of the revolving field are often used to force air upon the overhanging ends of the armature coil and thus cool the armature winding.

Protection of alternators from overloads. One of the most valuable characteristics of a well-designed alternator is its ability to withstand very severe overloads of momentary duration. It is often claimed that even on short circuit a first-class alternator cannot give sufficient current to cause it to burn out within a limited time, and it is without doubt true that the reactive drop in potential on full short circuit is so great that the current cannot under most conditions rise to a destructive value as instantaneously harmful to the windings as would be the case with a direct-current constant potential generator. At the same time, short circuits are greatly to be feared in alternating current plants, for the derangement of the system tends to upset the balancing of loads and distribution of potential to a serious extent, which may easily interrupt the service if continued for any length of time.

Modern switchboard practice tends to exclude every kind of automatic circuit-opening device between alternators and bus-bars. There is little doubt that this is wise, for experience in the last few years has definitely shown that the best place to protect a group of alternators is on the individual feeder or transmission circuits. A prominent New England engineer recently stated that in ten years' experience in several power plants with apparatus and switchboards he had never seen a case wherein an automatic device between generators and bus-bars would have been of any value, but, if used, would have been detrimental to the successful operation of the plants. The best practice locates the automatic switches in the feeder lines, so that a local short circuit will not trip alternator switches, for when several alternators are operated in parallel, tripping a circuit breaker on one tends to throw more load on the other

machines than they can readily carry instantly with good regulation, perhaps shutting down the whole station.

If the leads from the generators to the switchboard are carefully inspected and tested before installation, and run in conduit there need be practically no apprehension of trouble in this short distance. Probably 99 per cent of the electrical troubles in alternating-current systems occur in the transmission or distributing equipment, and it is a great advantage to be able to confine the fault to the local circuit upon which it occurs, either burning out the flow by the station capacity available or shutting down a single line while repairs are made. Careful operation of a station includes regular watching of the machine ammeters to see that the alternators are not continuously overloaded. Bus-bar short circuits are, of course, not guarded against by open circuiting devices in the outgoing lines, but these are of such rare occurrence that it would not pay to install automatic switches in anticipation of an overload from such a cause. There is no question that in an alternating plant it is better insurance to put the money which would otherwise be spent in expensive automatic oil switches and auxiliary wiring in the generator leads into increased generating capacity.

QUESTIONS

286. What is a dynamo?

Ans. A dynamo is a machine for transforming mechanical energy into electrical energy.

287. How are dynamos classified?

Ans. Into two grand divisions, viz., direct current dynamos and alternating current dynamos.

288. What is direct electrical current?

Ans. A current of unchanging direction.

289. What is an alternating current?

Ans. A current that reverses its direction of flow, periodically, from 20 times and upward per second.

290. Name the principal constituent parts of a dynamo?

Ans. The armature, the field, the collecting rings, or commutator, and the brushes.

291. How is electro motive force or current induced in a dynamo?

Ans. By rapidly changing field and armature relations by means of mechanical energy.

292. How is output of dynamo stated?

Ans. In Kilowatts equal to $1,000 \times \text{volts} \times \text{amperes}$.

293. How is output of motor stated?

Ans. In horse power, equal to $\text{Watts intake} \div 746$, $\times$ efficiency expressed decimally. (Not as a percentage.)

294. What is voltage of a dynamo? of motor?

Ans. It is the pressure the generator or alternator delivers at its own terminals. The voltage of motor is the voltage which should be applied to its terminals in order to develop full horse power.

295. What is full load current of dynamo? of motor?

Ans. Full load current of a dynamo is that current which may be drawn steady for 24 hours without causing any part of machine to exceed a safe temperature, i. e., 150° Fahr. This applies to factory motors.

Of a railway motor it is the current which passing through motor for one hour as it runs on the blocks in testing room, will cause it to rise to a temperature of 212° Fahr. We mean, of course, that the hottest part shall be no hotter than 212° Fahr.

296. What is meant by the rating of a dynamo? Of a motor?

Ans. The product of full load current multiplied by the voltage expressed in Kilowatts is rating of a dynamo. The actual mechanical horse power developed at the pinion of the motor as tested in shop. The gearing increases the power applied to and reduces the speed of the car wheels.

297. What is the armature core?

Ans. The sheet iron body which carries the armature winding and conducts the flux from pole piece to pole piece.

298. What is the armature spider?

Ans. The casting consisting of hub and arms which supports armature core.

299. What are binding wires?

Ans. They are narrow bands of phosphor bronze wire placed around armature every three or four inches to help bind winding to core. They rest on strips of mica and are sweated with solder all around.

300. What are commutator segments?

Ans. The commutator segments or bars are the copper pieces of which the commutator is built.

301. What are commutator leads?

Ans. They are the ends of the armature winding extending from core to the lug of the commutator bar.

302. What are pole pieces?

Ans. The end of magnet core nearest the armature. Usually larger than core.

303. What are magnet cores?

Ans. The iron inside the field coil.

304. What is the yoke?

Ans. The part of magnetic circuit connecting the magnet cores.

305. What is the pitch of an armature winding?

Ans. It is the number of teeth between the two sides of a formed coil plus one tooth.

Example: The two sides of a coil are in slots number 3 and 17, then pitch is 14.

306. Is there insulation between winding and core?

Ans. Yes. Mica or fuller board; there is also the tape on coil.

↓ 307. What insulation is there between conductors of winding?

Ans. The double cotton covering of each wire makes four thicknesses between conductors.

308. What is the air gap?

Ans. It is the air space between armature and pole pieces. In dynamos it is made as small as possible for efficiency.

In motors it is not made too small because this tends to make machine spark due to the weak field. In D. C. series motors it is from $\frac{1}{8}$ to $\frac{1}{4}$ of an inch, in A. C. series motor it is smaller, say $\frac{1}{10}$ to $\frac{1}{8}$ inch.

The larger the air gap of a motor the more the bear-

ings may wear before there is danger of armature rubbing against lower pole pieces.

309. What are field spools?

Ans. The brass shells on which the field coils are wound.

310. What is the commutator?

Ans. It is a series of copper bars placed parallel to shaft, insulated from each other and from the frame of machine. Each is connected to the winding and current flows from winding through them to brushes. It at the same time reverses the connections between the brushes and winding at the proper times so that the brush always collects current.

311. What is a collector or slip rings?

Ans. A collector consists of two or more rings of copper placed around the shaft and insulated from it, and each other. Each is connected to a part of the winding. The brushes rest on the rings.

They are used to collect current from a revolving armature style of alternator, to feed current into armatures of rotary converters, or the revolving fields of alternators.

The collector has no corrective influence and passes on the A. C. or D. C. current exactly as it receives it.

Single phase machines have two rings; two, three and six phase machines have three rings.

312. Is there a difference between no load and full load voltage of dynamos?

Ans. Yes. A shunt dynamo gives highest voltage at no load and lowest at overloads; the series dynamo gives lowest at no load and highest at full load. The compound dynamo is a combination of series and shunt, and gives some voltage at all loads.

An alternator acts like a shunt dynamo.

313. What is a field rheostat?

Ans. It is a resistance in the field circuit which can be varied to change the current, and hence the field strength. This alters the voltage of dynamo.

314. What are commutated fields?

Ans. In some motors the field coils are arranged in sections so that they may be arranged in parallel or series, or in combinations.

All coils in parallel give the greatest current and hence slowest speed of motor; all coils in series give the weakest field and the fastest speed.

315. What relation has field strength to speed of motor?

Ans. The weaker the field the faster the speed, for the motor must revolve fast to generate its proper counter E. M. F.

316. What relation has armature strength to speed of motor?

Ans. The greater the armature current the greater the speed.

317. What effect on the power of motor does field and armature strength have?

Ans. The greater the field and armature current the greater the power.

318. What is a ring winding?

Ans. One which passes over and under around the core, a space being left between shaft and core to accommodate winding.

319. What is a drum winding?

Ans. One where all winding is on the outer surface of core.

320. Upon what does sparkless commutation of current depend?

Ans. (1) The more commutator bars the better, there being less voltage and therefore tendency to spark between bars. The average railway motor has from 100 to 125 bars on commutator.

(2) The fewer the ampere turns on the armature in comparison to the ampere turns on the field the less sparking.

(3) The more turns short circuited by the brush when touching two or more bars at once the greater the tendency to spark.

321. What is a shunt field?

Ans. One whose coils are placed as a shunt across the brushes. It carries a small current.

322. What is a series field?

Ans. One which carries the main or nearly all the main current and is placed in series with the armature. A small strip of resistance metal is used sometimes to divert a portion of the main current from the series field.

SWITCHBOARDS.

Switchboards are made up of panels of slate on a frame of angle iron. Each panel is designed for certain work so that a description of the different kinds of panels is sufficient.

The first board to consider is the D. C. outgoing line board, served from D. C. generators.

D. C. GENERATOR PANELS.

Fig. 281 shows three generator panels each of which is regularly equipped, from a capacity of 250 to 6,500 amperes with;

1 Carbon break or magnetic blow-out circuit breaker, with telltale.

1 Illuminated dial ammeter with shunt.

1 Hand wheel and chain for operating rheostat.

1 Receptacle for voltmeter plug.

1 S. P.-S T. field switch.*

1 S. P.-S. T. main switch.

1 Recording Watt-hour meter.

A rear view of these panels is shown in Fig. 282.

The best practice puts a main switch at the machine

*S. T. means single throw.

D. T. means double throw, i. e. the switch has two sets of clips and can be thrown into either of them.

S. P. means single pole.

D. P. means double pole, i. e. opens both sides of circuit.

T. P. means triple pole, i. e. opens every conductor of a 3 phase system.

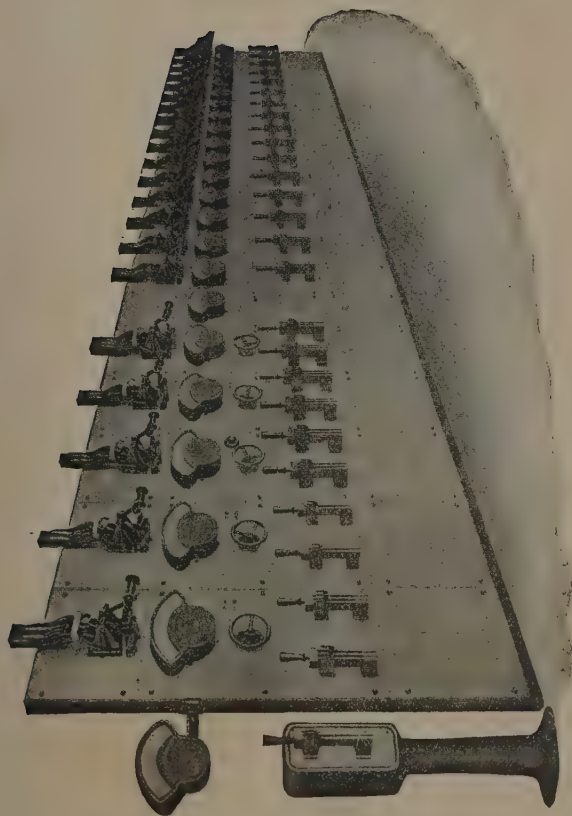


Fig. 279.

so that the cables from machines to board may be cut off from generator. It is also good practice to run the equalizer cable along in ducts from machine to machine without carrying it to the board.

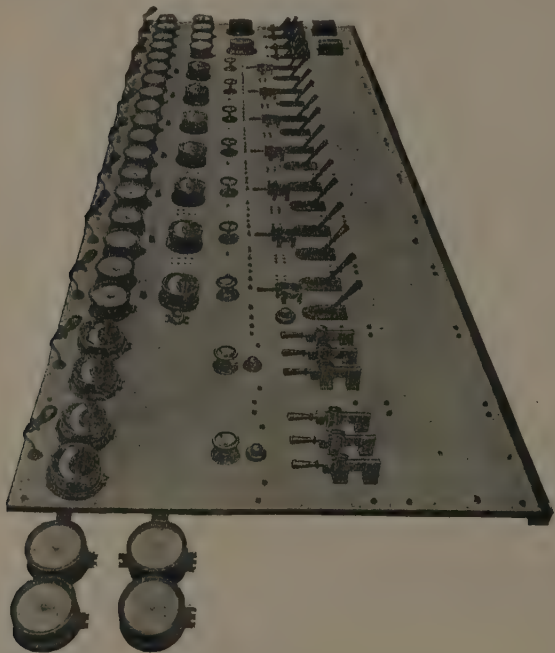


Fig. 280.

This equalizer connects the junctions of series field and brush on all machines as shown in Fig. 283; the shunt coils being omitted to simplify diagram.

It is best to place the main switch and equalizer switch on a pedestal panel as shown in Fig. 284 for moderate



Fig. 281. D. C. Generator Panels.



Fig. 282. Rear View of Fig. 281.

capacity and in Fig. 285 for 4,000 ampere (and larger) machines. The upper switch being the main switch. The rear view of these large capacity pedestals is shown in Fig. 286.

A better view of the 4,000 ampere toggle operated main switch is given in Fig. 287. The quick-break S. P.-S. T. switch is illustrated in Fig. 288.

The field switch, Fig. 289, has a carbon break. Just before the switch opens it makes contact with an extra clip which puts a resistance on as a shunt around the field coils.

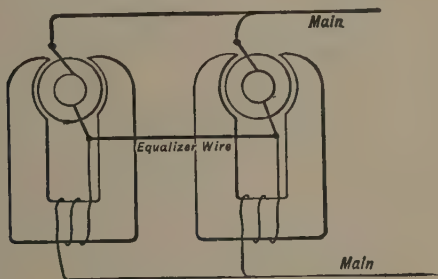


Fig. 283. Equalizer.

If this were not done the fields would act like a kicking or spark coil and their insulation be damaged.

In Fig. 290 is seen the diagram of the panel shown in Figs. 281-282 when capacity is 800 K. W. or under.

Fig. 291 shows the same panel when capacity is larger. The panel at left is for 1,000 and 1,200 K. W., the next for 1,500 K. W. and over. The cuts on right side show the back and side view of the 1,500 K. W. panel.

The scheme of electrical connections for panel of Fig. 281 is shown in Fig. 292.



Fig. 284. Pedestal Panel for Main and Equalizer Switches.
Small Capacity.



Fig. 285. Main and Equalizer Switches for Large Capacity.

D. C. FEEDER PANELS.

A set of feeder panels for one feeder each is shown in Figs. 293 and 294, a panel for two feeders with separate switches and one ammeter reading sum of both currents is shown in Fig. 295, while Fig. 296 has an instrument and switch for each circuit.

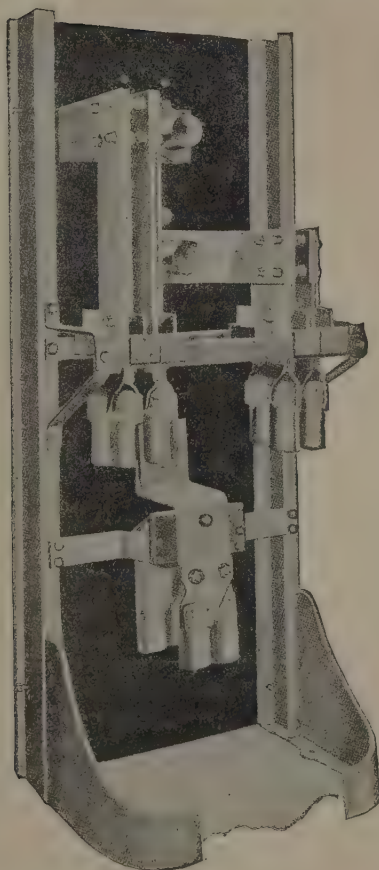


Fig. 286. Rear View of Fig. 285.

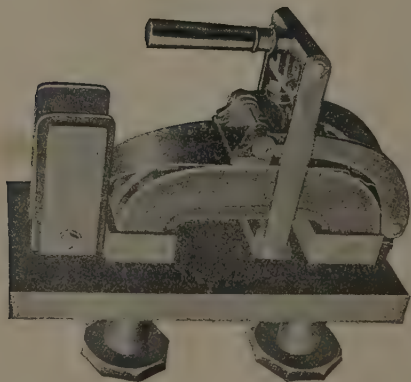


Fig. 287. 4,000 Ampere Toggle Operated Switch. Laminated Main Contact, Carbon Secondary Contact with Magnetic Blowout.



Fig. 288. 3,600 Ampere Quick Break Switch.



Fig. 289. Field Discharge Switch.

Fig. 297 gives the diagram of these feeder panels and Fig. 298 gives the electrical connections.

With panels as described the way to throw a generator in parallel with other generators already running, the following procedure should be followed:

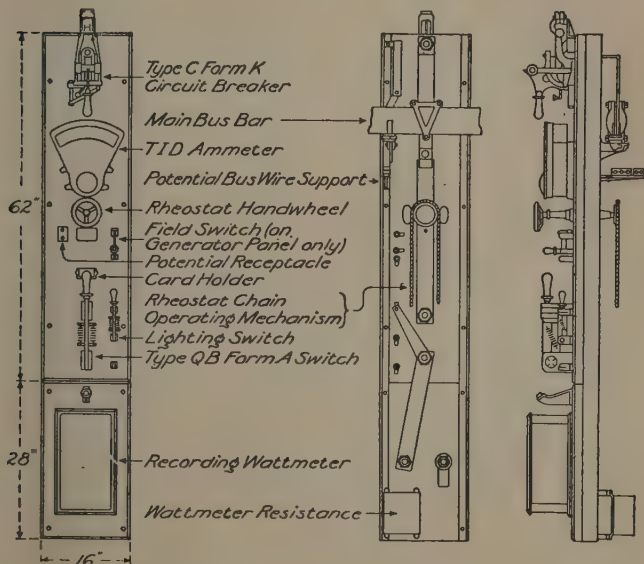


Fig. 290. Construction of Fig. 281 for Small Capacity.

First—Close main and equalizer switches (on pedestal or panel near machine).

Second—Close field switch (on panel).

Third—Close circuit breaker.

Fourth—Insert potential plug in receptacle and regulate voltage.

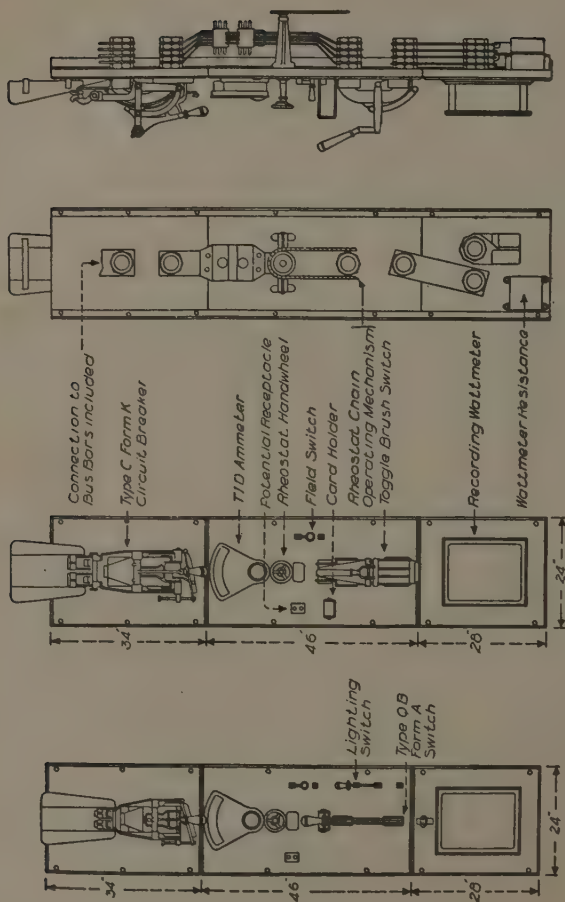


Fig. 291. Construction of Fig. 281 for Large Capacity.

Fifth—When the proper voltage is obtained, close the other main switch (on panel).

All the above applies to the distribution of the output of rotary converters, but as they have some peculiarities they will be considered later.

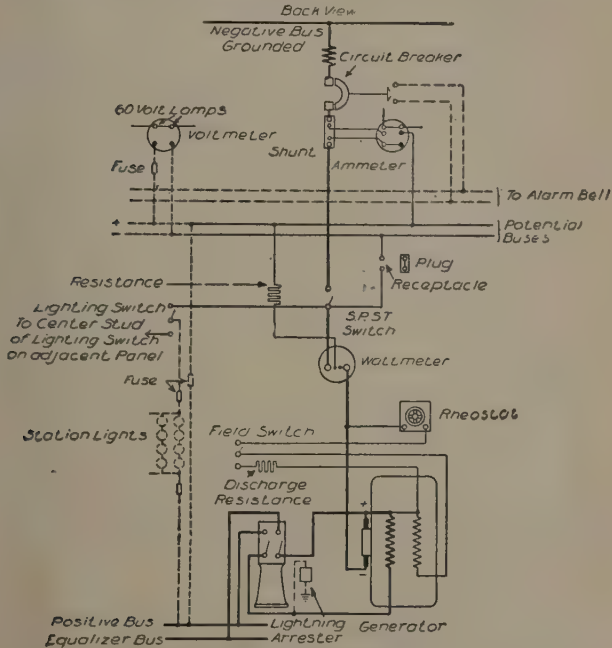


Fig. 292. D. C. Generator Panels.

Nearly all railroads are operated by a 3 phase station although the trains are run by single phase A. C. or by converted power as D. C.

The consideration of 3 phase A. C. switchboards will cover all the A. C. power houses.



Fig. 293. D. C. Railway Feeder Panels.

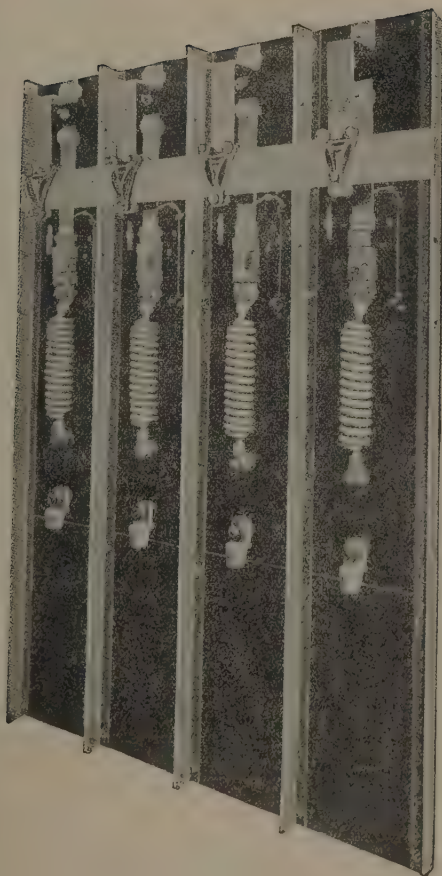


Fig. 294. Rear View of Fig. 293.



Fig. 295. Two Feeder D. C. Panel.



Fig. 296. 1,200 D. C. Ampere, Railway Feeder Panel for Two Circuits.

A. C. GENERATOR PANEL.

The panel in Fig. 299 contains:

1 Horizontal edgewise balanced three-phase indicating wattmeter, arranged for reading both the kilowatts output and the wattless component.

1 Horizontal edgewise ammeter.

1 Horizontal edgewise voltmeter.

1 Balanced three-phase induction recording wattmeter.

1 D. P. D. T. potential reversing switch for the indicating wattmeter.

1 Four-point receptacle for synchronizing connections.

1 Hand-wheel and chain operating mechanism for field rheostat.

1 S. P. S. T. carbon break field switch with discharge clips.

1 D. P. D. T. engine governor control switch.

1 T. P. S. T. oil switch.

1 Current transformer for instruments.

2 Potential transformers for instruments.

The functions of the instruments are to indicate the current, voltage and kilowatts output of the generator, and the wattless component of the output. For indicating the wattless component, the potential coil of the indicating wattmeter is wired to the potential reversing switch, which is normally held by a spring so as to connect the instrument up as a wattmeter. By throwing the switch against the spring into the other position the potential coil is reversed and the instrument reads the wattless component, giving a ready means of detecting any currents flowing between the alternators which are operating in parallel.

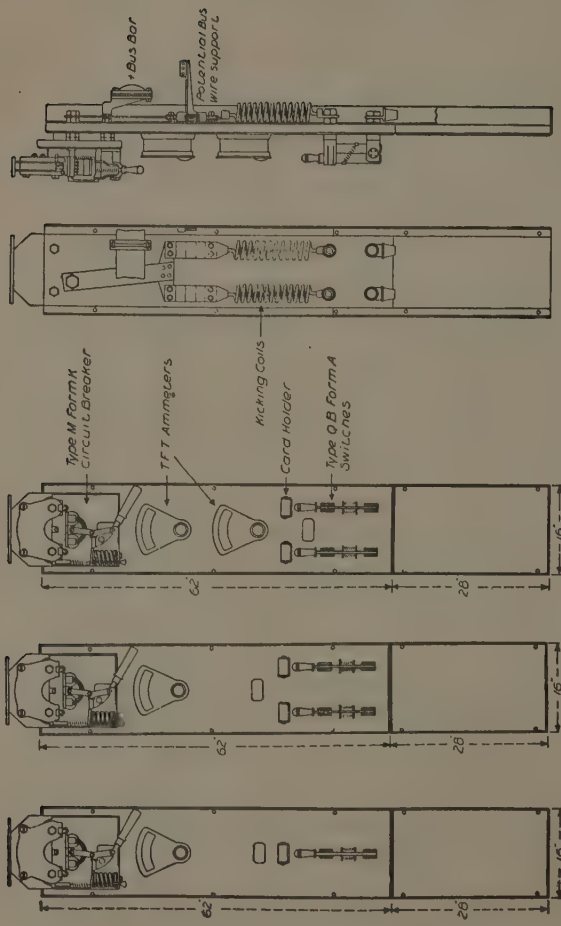


Fig. 297. Construction of Figs. 293 and 296.

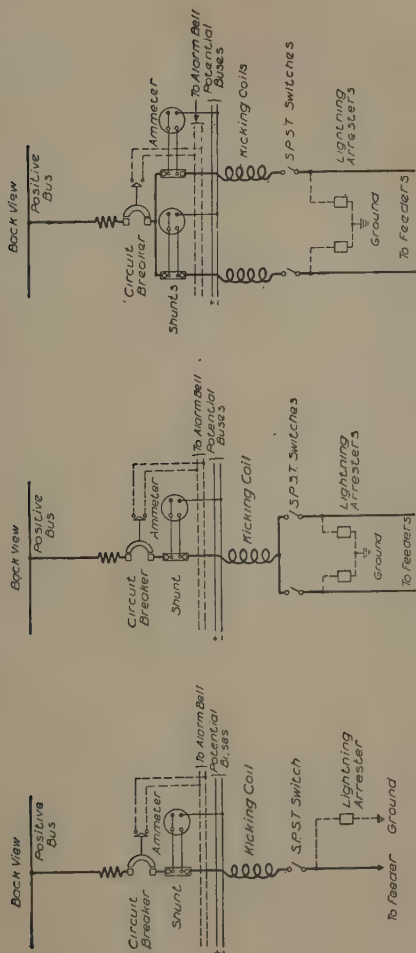


Fig. 298. Three Styles of D. C. Feeder Panels.



Fig. 299.
A. C. Generator
Panel.

The engine governor switch is to operate the motor which temporarily controls the governor on engine, or turbine when their speeds are being altered to bring two alternators into synchronism or adjusting the division of load when operating in parallel.

The generator oil switch has no automatic overload release, as it is important to keep the generator in service during heavy short circuits caused by trouble on the transmission lines. When such short circuits occur, the generators are immediately relieved by the opening of the automatic line switches.

The diagrams for connecting up generator panels according as transformers are, or are not used will be found in Figs. 300 and 301.

A. C. OUTGOING PANEL.

The panel on left of Fig. 302 contains:
3 Horizontal edgewise ammeters.

1 T. P. S. T. oil switch, with overload release.

3 Current transformers.

Three ammeters—one for each phase—are furnished for each line, to facilitate the detection of unbalancing due to open circuits or leakage. With balanced loads, the ammeter pointers should show equal deflections under normal conditions. As the ammeters are arranged in a

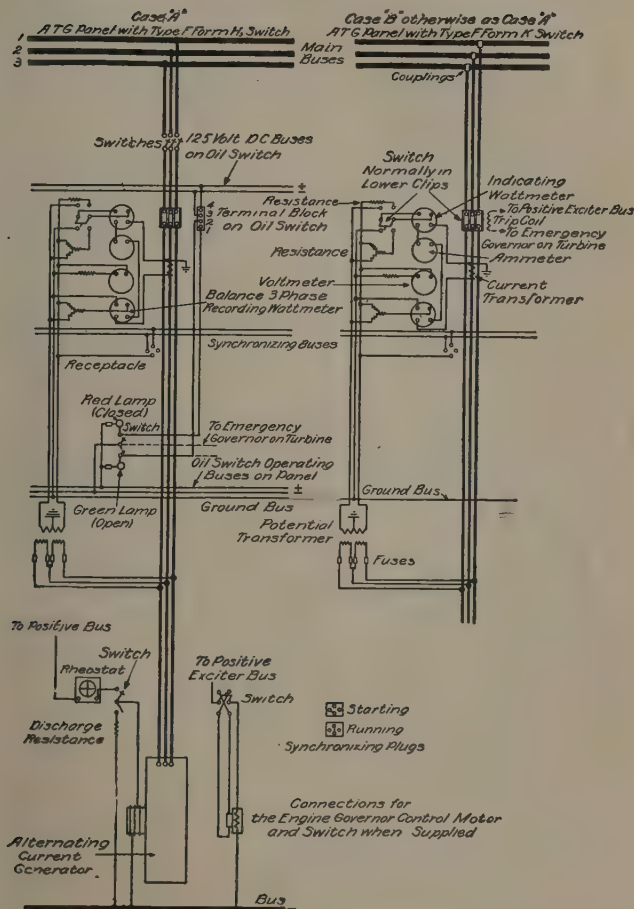
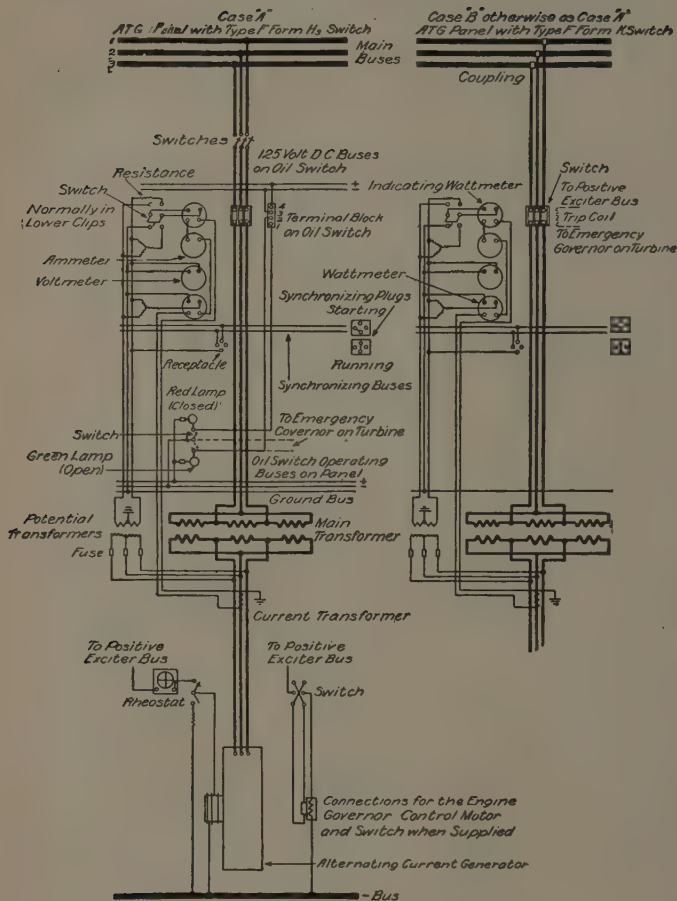


Fig. 300. A. C. Generator Panel without Step-up Transformers.



ALTERNATING CURRENT GENERATOR PANEL FOR GENERATOR WITH STEP-UP TRANSFORMER

Fig. 301. A. C. Generator Panel for Generator with Step-up Transformer.



Fig. 302. A. C. Outgoing Line Panels.

perpendicular row any variation in the deflection of the pointers is readily detected.

The current transformers serve to operate the ammeters and the automatic release on oil switches.

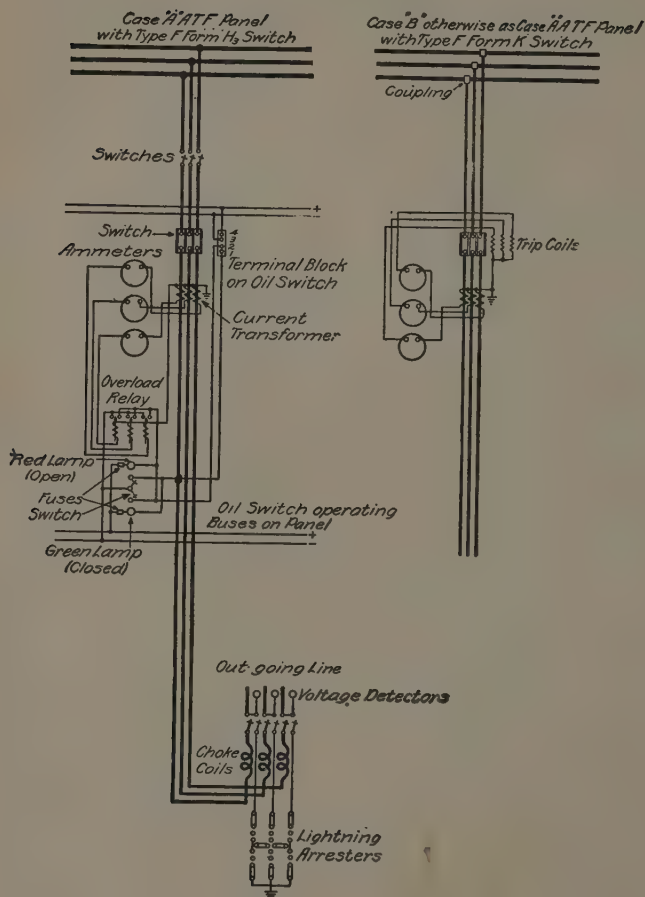


Fig. 303. A. C. Outgoing Line Panel.

The panel on right of Fig. 302 has but one ammeter and merely has the handle for operating the oil switch. The actual switch being in a brick compartment at rear of panel. The overload relay (3 pole) which trips the oil switch is at base of panel.

Fig. 303 gives the electrical connections of panels in Fig. 302.

The swinging bracket of Fig. 304 contains a synchronism indicator, two lamps for synchronizing (practically a duplicate set of synchronizers) and a voltmeter for the station exciter generator.*



Fig. 304. Synchronism Indicator and Exciter Voltmeter on Swinging Bracket.

To use the synchronism indicator put one plug in on panel of a generator which is running, and the other plug in the panel of the generator which is starting.

Fig. 305 shows a complete switch board of one generator panel in center, a panel for one outgoing line on the right, an exciter panel on left, with the swinging bracket on extreme left.

*D. C. Generator furnishing current for fields of alternators.



Fig. 305. Main Station Switchboard for One A. C. Generator and One Outgoing Line.

Such a switch-board would be extended towards the right indefinitely, as more lines were put on the station, by the addition of more outgoing line panels.



Fig. 306. Exciter
Panel Auxiliary
Lighting Switches
on Sub-Base.

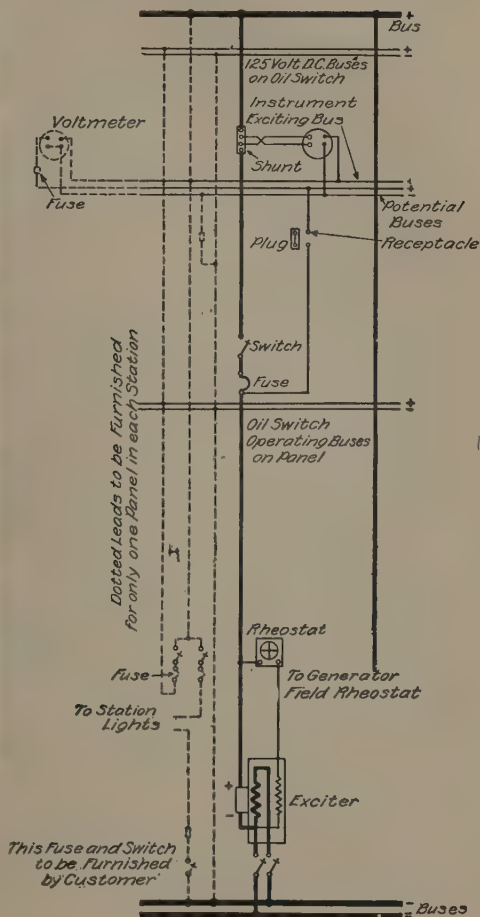


Fig. 307. Exciter Panel.

EXCITER PANEL.

Each exciter panel is equipped with:

- 1 Thomson feeder type ammeter.
- 1 Hand-wheel for operating rheostat.
- 1 Two-point potential receptacle connected to voltmeter.
- 1 S. P. S. T. positive lever switch, with fuse mounted back of panel.

One Exciter Panel in every switch board is furnished with the following additional switches: (as in Fig. 306.)

- 2 S. P. S. T. lever switches, with fuses back of panel, for the control of station lighting and auxiliary circuits.

On the frame of each exciter there are required the following switches, mounted on a common slate base:

- 1 S. P. S. T. negative lever switch.
- 1 S. P. S. T. lever switch for equalizing.

The exciter panels are designed single pole, *i. e.*, only the positive leads of the generators are connected to the switch-board panels and only the positive bus-bar is mounted back of them. The negative and equalizer leads are connected through their switches to the negative and equalizer bus-bar, which are placed under the floor near the exciters. With the bus-bars of opposite polarity so widely separated there is practically no chance of short circuit of the exciter connections. The positive field leads of the alternators are carried to the panels, while the negative field leads are permanently connected to the negative exciter bus-bar.

Fig. 307 will give the electrical connections of an exciter panel.

The blower motors running the blowers which cool transformers are of the 3 phase induction type, or D. C. shunt motors.

The D. C. motors are started by the regular starting box, Fig. 308.



Fig. 308. Starting Panel
for D. C. Blower Set.



Fig. 309. Main Switch Panel
for A. C. Blower Set.

The current to an induction motor is controlled by a switch like Fig. 309, if from auxiliary low voltage buses, or from an oil switch on a panel like Fig. 310, if full station voltage is used.

The actual starting is done by a switch as in Fig. 311, which is between secondaries of transformers or reactance coils and the induction motor.



Fig. 310. Oil Switch A. C.
Panel for Incoming Line
Motor Driving Exciter or A. C.
Side of Rotary.



Fig. 311. Induction Motor
or Rotary Starting Panel.

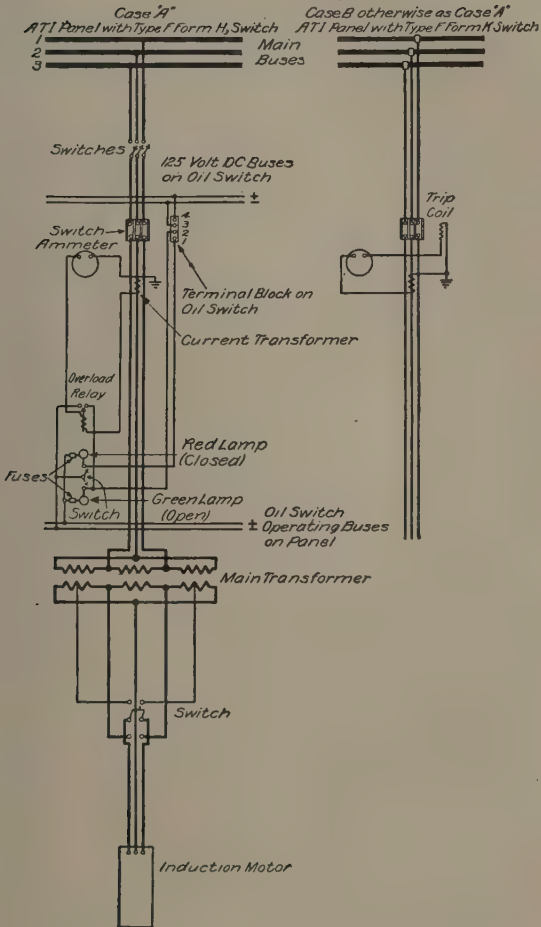


Fig. 312. Induction Motor Panel.

Fig. 312 shows connections of an induction motor to main buses, using an oil switch and a starting switch.

The operation of several sub-stations on a single line is generally recognized as good practice, especially for interurban systems on which the traffic is not very dense.

To insure continuity of service in the event of line trouble, it is expedient to sectionalize the line at every sub-station that is located at an intermediate point of the line. This sectionalizing is accomplished at each intermediate station by carrying the incoming line to the bus-bars through the air brake disconnecting switches which are installed in connection with the arresters, and by carrying the outgoing line through an oil switch. In case of line trouble, this arrangement allows all sections of the line between the generating station, and any section on which the trouble occurs to be operated continuously. The power is automatically cut off from the section in trouble by an oil switch in the outgoing line panel equipment of the sub-station at the generating station end of the section, so that the air brake disconnecting switches in the sub-station at the other end of the section need never be opened under load.

When duplicate transmission lines are used, two incoming line panels and two outgoing line panels are recommended for each intermediate sub-station. The installation of these individual panels facilitates the disconnection of either line of any section and the continuance of the service over the other line of the section without any interruption

SUB-STATIONS AND SUB-STATION SWITCH-BOARDS.

The sub-station is the connecting link between the transmission line, and the trolley wire, or third rail.

With A. C. generation and D. C. utilization of power the sub-station with its rotary converters is a necessity. Where A. C. is used straight through, the sub-station may be omitted, by having each station fed directly, with a very high tension trolley wire.

Usually there will be sub-stations containing only transformers. The 22,000 volt lines will feed these transformers and they will feed the line with some lower voltage. The scheme of placing costly generators directly on line, running the danger of static and lightning discharges, seems rather rash.

When transformers and sub-stations are used the generators may deliver a much higher voltage, for the transformers step it down later, and the absence of any electrical connection between primary and secondary protects the generator by absolutely separating it and the trolley wire.

It may be said that in A. C. work each car or locomotive carries a small sub-station, *i. e.*, a transformer to lower the line voltage to a pressure suitable for the motors.

THE A. C. SUB-STATION.

One of the great advantages of A. C. traction is the simplicity of its sub-stations. They are small brick buildings containing the transformers for lowering voltage, a simple in-coming and out-going switchboard.

The best frequency for motors is a low one, 25 being now standard. It has been recently proposed to design systems at a frequency of 15.

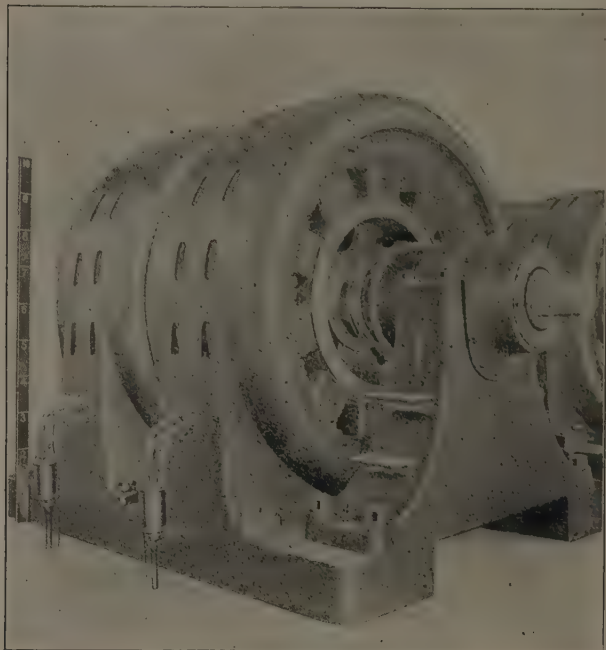


Fig. 313. A Frequency Changer.

Both of these frequencies are rather low for lighting and transformers are cheaper to build and operate as the frequency increases.

Railroads by purchase often acquire electric roads which as a side issue furnish light to some municipality.

They may continue to do this at the desired frequency from their low frequency plant by installing a frequency changer, Fig. 313.

This consists of two similar machines, one a simple A. C. motor (called a synchronous motor) and the other an A. C. generator built much like an induction motor. It takes in current of one frequency and delivers current at another frequency, higher or lower as desired.

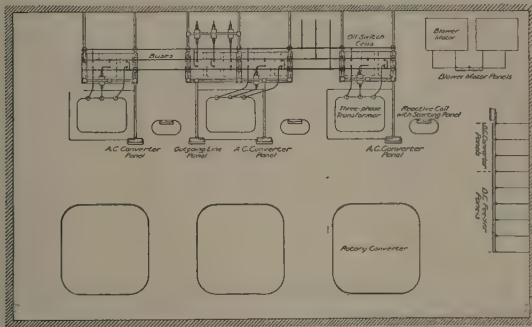


Fig. 314. Floor Plan of a Typical Sub-Station.

The amount the frequency is changed, is determined by the design of the machine, and is fixed once for all. There can be no regulation of the frequency, or in fact is that desired; all that is needed is a change from 25 to say 133 and have it keep at 133.

THE A. C. TO D. C. SUB-STATION.

When D. C. third rail or trolley is used the sub-station contains the step-down transformers which feed the rotary converters, the converters themselves, and the incoming A. C. and out-going D. C. switchboards.

The layout of a typical sub-station is shown in Figs. 314-315-316. The first gives a plan of the station showing position of each piece of apparatus on the floor. The

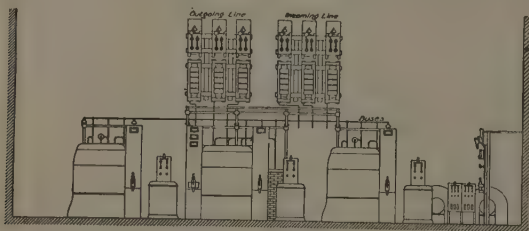


Fig. 315. View of Rear of Station Shown in Fig. 314.

second gives view of the transformers with an A. C. panel for each rotary along side of transformer on right. On left of middle transformer is the panel for outgoing A. C., which goes to next sub-station.

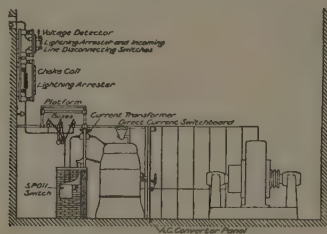


Fig. 316. End View of Fig. 314.

The starting resistances and switch for the rotaries are between the transformers, and between the last transformer and blower panels.

High up on the wall are the two lightning arresters and fuses for the incoming and outgoing A. C. lines.

Fig. 316 shows an endwise view of the station which explains itself.

The sub-station shown in Fig. 317 has one complete unit in full view. It consists of a 1,000 K. W., 6 phase rotary fed by a set of 3 single phase transformers connected so as to make one 3 phase transformer. These are air blast type.



Fig. 317. Sub-Station of 6 Phase Rotaries. Surface Roads, New York.

At the extreme left is an induction regulator operated by a small motor on its top. This motor is controlled from the switchboard. The rotaries in this station are started from the D. C. side, because at least one rotary is always running.

Fig. 319 is an interior view of a sub-station showing a rotary converter, switchboard and the transformers in the rear.

SYNCHRONOUS MOTOR.

If two similar single phase machines are connected to a line the one may be driven as a generator and the other considered as a motor. This motor will not start when its switches are closed, because the rapid changes of the armature polarity give impulses to move first in one direction and then in another; with the result that motor stands still.



Fig. 318. Sub-Station, Rotaries with Starting Motors, Pittsfield, Mass.

If the motor machine should be revolved by a smaller motor till it is moving at the same speed as generator, or slightly faster, when the switches of the motor are thrown the motor will continue to revolve at generator speed.

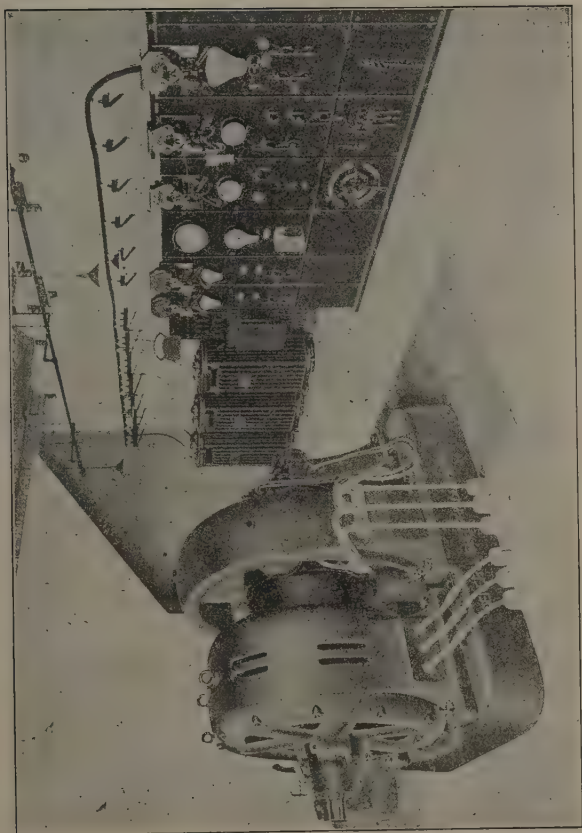


Fig. 319. Sub-Station.

Such a motor will carry any load up to 50% overload without the slightest change in speed until suddenly it stops. To get it started again the load must be thrown off and the motor brought up to its proper speed by the auxiliary motor.

The D. C. for the motor field is supplied to the motor from external source. The strength of the field has no influence on the speed of motor.

The speed is governed by the frequency of generator. If the generator passes 25 pairs of poles a second then the motor will do the same. If the generator and motor have the same number of poles, then they revolve the same number of revolutions per minute. A 32 pole generator driving a 16 pole motor goes only half as fast as the motor, for if frequency of generator is 25 the motor must go twice as fast so as to pass the same number of pairs of poles a second.

INDUCTION REGULATION.

If the current going to a rotary passed through an auto transformer where part of the coil could be placed in opposition to the rest of it, then we would have an arrangement capable of slightly lowering or raising the voltage applied to rotary, according as the part of coil was in series or in opposition to the rest of the coil.

The *induction regulator* produces the same effect in a little different way. It is as shown in Fig. 320. It consists of a primary P. Q. of a transformer across the line as a shunt and a secondary S. T. in series with one of the mains.

An iron core E. F. when in this position causes the flux to pass through secondary and induce an E. M. F.,

which is added to the line, while if turned by the handle or by motor into position H. G. the flux through S. T. is reversed and the E. M. F. induced is subtracted from the line. This transformer carries full current, but induces so few volts that the power is small. When only 3 or 4% of its own power is lost, it is a very small fraction of total power delivered to rotaries.

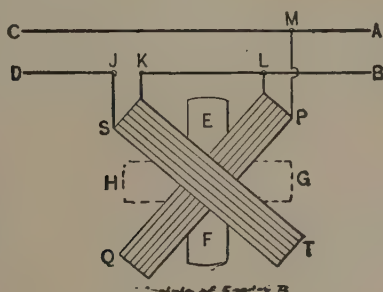


Fig. 320. Induction Regulator.

STARTING ROTARY CONVERTERS.

To start from the A. C. mains as a polyphase motor the connections shown in Fig. 321 are used.

The line in this case has a compensator attached and as shown in Fig. 321 the rotary will now receive only a fraction of the line voltage, and starts at slow speed.

As the taps to rotary are moved up the coils, the voltage is higher until finally the coils of compensator are cut out entirely, and rotary is revolving at full speed, taking full line voltage.

When transformers are between line and rotary the

same scheme is applied to the secondaries of the transformer.

To start from D. C. mains a regular starting set is used and operation is like any shunt motor.

SYNCHRONIZING.

When a rotary is up to speed and ready to be connected to the A. C. line, it must be connected at just the right instant.

This time can be determined by bridging across the connecting switch, some lamps in series.

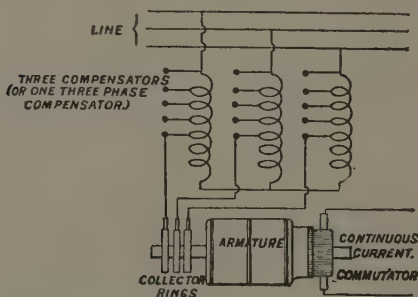


Fig. 321. Starting 3 Phase Rotary with Compensator.

Both the rotary and the incoming line will be operating these lamps. As a result they will flicker. As the right instant is approached the lamps blink slowly until the change from darkness to light takes one or two seconds; then at the time of maximum brightness or darkness close the connecting switch. The lamps may then be disconnected.

Two voltmeters can be combined in one case and made to indicate through a single pointer. When the line and rotary agree the pointer stands in center of dial. These are called synchronism indications or synchronizers.

By synchronism for two alternators we mean that the maximum value of the E. M. F. in each machine occurs at exactly the same instant. They are then in phase, and in step and in synchronism. In this condition they run as generators in parallel.

If they get out of phase sufficiently they will keep in step and in synchronism but will be in opposition and one will continue to generate while the other will begin to absorb power and act as a motor. This is what is meant by synchronism for an alternator and a rotary converter.

SUB-STATION SWITCHBOARDS.

The arrangement of switchboard panels in sub-stations differs from the generating station arrangement, in that the alternating current panels are, preferably, installed individually near their several line, or transformer oil switches, thus greatly simplifying the high tension wiring and the oil switch operating mechanism, while the continuous current panels are grouped into a single switchboard conveniently located. The location of the alternating current panels, transformers, oil switch cells, continuous current switchboard, and machines, as shown, in Figs. 314, 315 and 316, is suitable for a 13,200 volt 3 phase sub-station.

Standard sub-station line panels are equipped as follows:

A. C. INCOMING LINE PANEL.

These are only used when lines are in duplicate, and are shown in Fig. 310.

A. C. OUTGOING LINE PANEL.

These are necessary in every sub-station except the terminal one. This panel is shown on left of Fig. 302.

The ammeter on incoming panel must be able to show all the current ever drawn by its own sub-station and those beyond. The capacity of the three ammeters on any outgoing line panel, however, need only be large enough to read the maximum load on the line beyond the sub-station in which the panel is installed. As already stated for the outgoing panels of the generating station, three ammeters are furnished in order to indicate, by their unbalanced readings, any open circuit leakage on the outgoing line.

A. C. ROTARY CONVERTER PANEL.

The panel controlling the intake of the rotary is shown in Fig. 302 (on right). It contains:

- 1 horizontal edgewise ammeter.
- 1 T. P. S. T. overload relay for oil switch.
- 1 current transformer.

THREE-PHASE ROTARY STARTING PANELS.

In Fig. 311 is shown this panel. The switch is used for starting the converter by connecting two of the three converter leads, first to half-voltage taps and then to the full voltage terminals of the transformer. The third converter lead is permanently connected to the third terminal of the transformer.

SIX-PHASE ROTARY STARTING PANELS.

As shown in Fig. 322 contains 2 T. P. D. T. lever switches.

These switches are used for starting the converter by connecting three of the six converter leads, first to one-third voltage taps, next to two-thirds voltage taps, and finally to the full voltage terminals of the transformer. The other three converter leads are permanently connected to the remaining three transformer terminals.



Fig. 322. Six-Phase Rotary Starting Panel.

POSITION OF PANELS.

The A. C. incoming line panels are connected between the high tension bus-bars, and the three-phase primaries of the step-down transformers, and are, therefore, suitable either for three or six-phase converters. Converters of capacities up to 400 kilowatts are usually three-phase, and above that capacity, six-phase. The secondaries of

the transformers are not carried to the main switchboard, but are connected through separately mounted starting panels to the converters. The dimensions of rotary converter starting panels vary with the capacity of the converters, but they are of suitable size for mounting near the latter. In most cases where compound wound converters with reactive coils are used, the coils are connected between the transformer secondaries, and the converters, and the starting panels are mounted on top of the reactive coil casings.

ROTARY OUTPUT PANELS.

These panels are the same as D. C. generator output panels and are shown in Fig. 323.

The circuit breaker is furnished with a low voltage release coil, which trips when the voltage drops to approximately half of its normal value, and with a tell-tale switch, for connection to an alarm bell which will ring when the breaker opens automatically. To prevent racing of the converter caused by accidental disconnection on the alternating current side, an adjustable speed limiting switch is mounted on the converter shaft. This speed limiting switch operates to short circuit the low voltage release coil when the speed of the converter increases a certain degree above normal, thereby tripping the breaker and disconnecting the converter from the continuous current bus.

The continuous current rotary converter panel is connected in the positive side of the circuit, and the negative side is permanently connected to the grounded negative bus, which is run along below the machine. The equalizing switch, and the switch for opening the shunt to the



Fig. 323. Rotary
Output Panel.

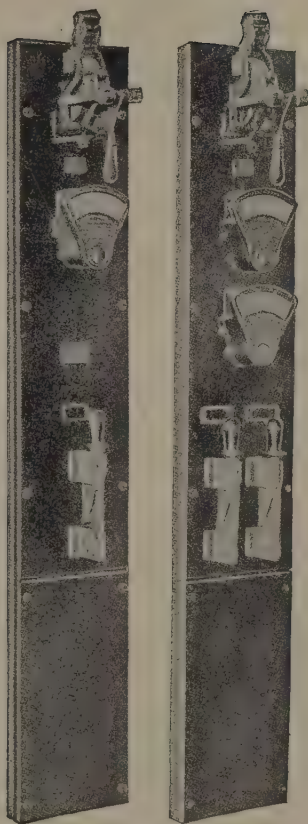


Fig. 324. D. C. Feeder Panels.

series field, required for a compound wound rotary converter, are mounted directly on the frame of the machine.

The advantages of equalizing on negative side, and of using single pole switches has been discussed under generators.

D. C. OUTGOING LINE PANELS.

The panels for one and two circuits are shown in Fig. 324.

PLACING ROTARIES IN SERVICE.

After a rotary converter has been started from the alternating current ends, and builds up with the proper polarity, the following procedure should be followed, to throw the direct current end in parallel with other machines running:

First—Close equalizer switch (on machine).

Second—Close circuit breaker (on panel).

Third—Insert potential plug in receptacle and regulate voltage.

Fourth—When the proper voltage is obtained close positive switch (on panel).

SWINGING BRACKET FOR VOLTMETER.

As shown in Fig. 325 a swinging bracket is provided to be mounted at end of board, carrying:

1 Thomson illuminated dial station type D. C. voltmeter.

1 plug for insertion in receptacles on the rotary output panels, thus connecting the voltmeter to the particular machine.

If the rotary when started from A. C. side comes up with polarity reversed, the voltmeter will swing back of zero.

The polarity of the converter can be corrected by operating the four pole double throw *field break-up reversing switch*, which is mounted on the frame of the converter and included in its equipment. This field reversing switch stands normally in its "up" position, and the op-



Fig. 325. Voltmeter Bracket.

eration of throwing it into the "down" position reverses the field magnetization. When the voltmeter needle indicates that the throwing of the switch into the "down" position has established correct polarity, the switch should immediately be thrown back into the "up" position, thereby connecting the field correctly for service.

The operation just described should be performed while the alternating current side of the converter is connected to the low voltage taps of the transformer. When the voltmeter indicates the correct polarity, the converter may be connected to the full voltage terminals

of the transformer. The continuous current voltage may then be adjusted by means of the field rheostat, and the machine thrown on the continuous current bus.

When motor operated oil switches are installed in the sub-station, a *storage battery* is used as a source of power for the small 110-volt series motors which operate the switches.



Fig. 326. Storage Battery Panel.

The storage battery consists of :

55 cells, giving 110 volts, the type and capacity of the cells depending upon the number of switches to be operated.

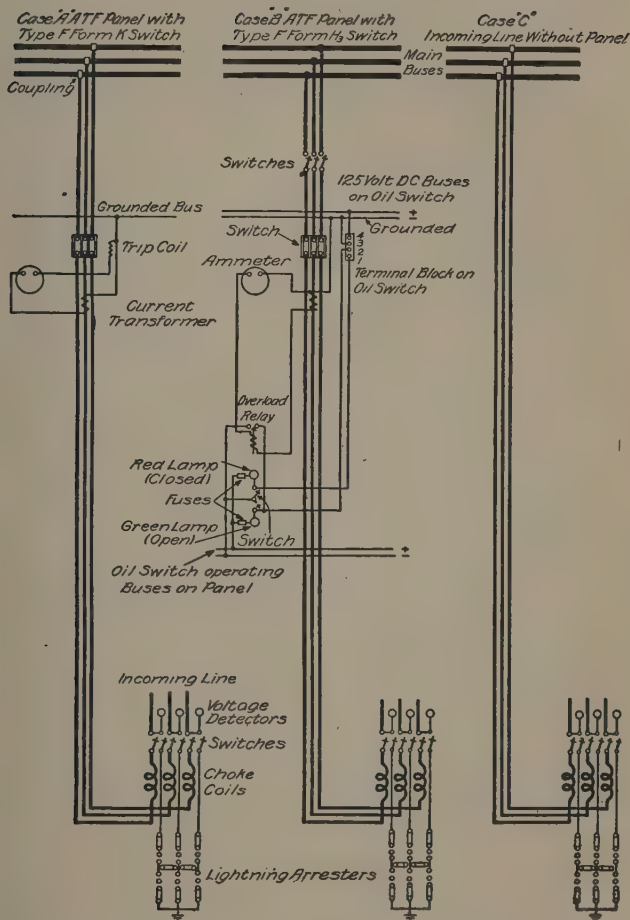


Fig. 327. Outgoing A. C. Line Panel.

STORAGE BATTERY PANEL.

As shown in Fig. 326 contains:

1 Thomson feeder type ammeter, with zero at center of scale.

1 Rheostat.

2 S. P. S. T. quick-break switches.

2 enclosed fuses.

The battery is charged by connecting it in series with the rheostat across the 600 volt bus-bars. The rheostat is provided with a dial switch for regulating the charging current.

BLOWER MOTOR PANELS

Are required for controlling the induction motor blower sets, which are used with air blast transformers. Each panel, as shown in Fig. 309, contains:

1 T. P. D. T. lever switch.

3 enclosed fuses.

The motors are usually wound for 350 volts, and are operated from the secondary side of the power transformers. The panels are located near the blowers.

As the transformers are not connected in parallel on the low tension side, the double-throw switch is furnished for connecting either of the duplicate blower motors, which are usually furnished, to whichever bank of transformers happens to be in use, as already described for the generating station.

ELECTRICAL CONNECTIONS OF PANELS.

A. C. incoming lines are shown in Fig. 327.

Connections for a three-phase rotary high voltage panel, low voltage starting panel and blower motor panel

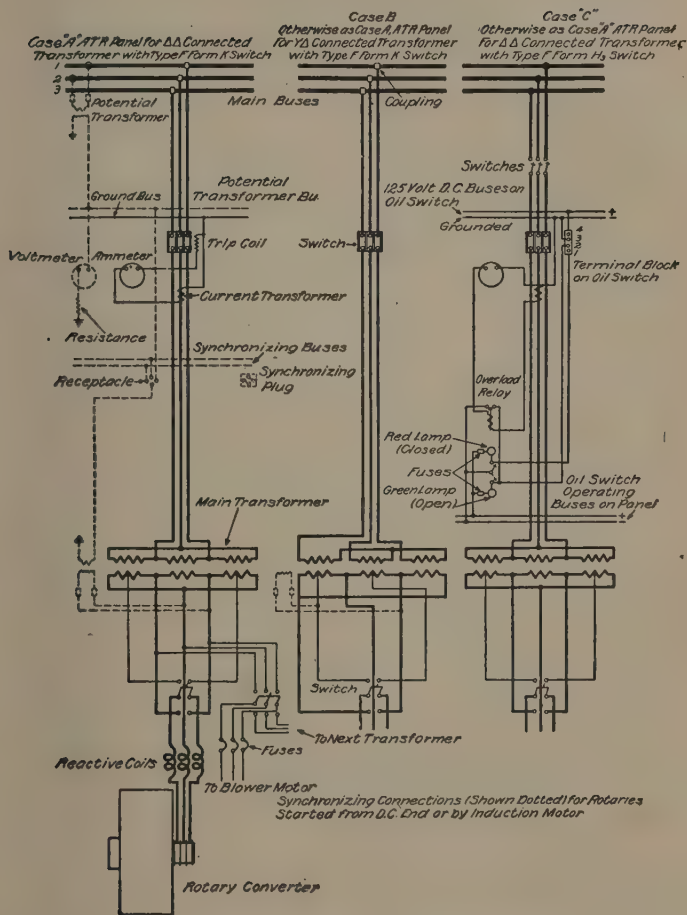


Fig. 328. A. C. End of Three-Phase Rotary.

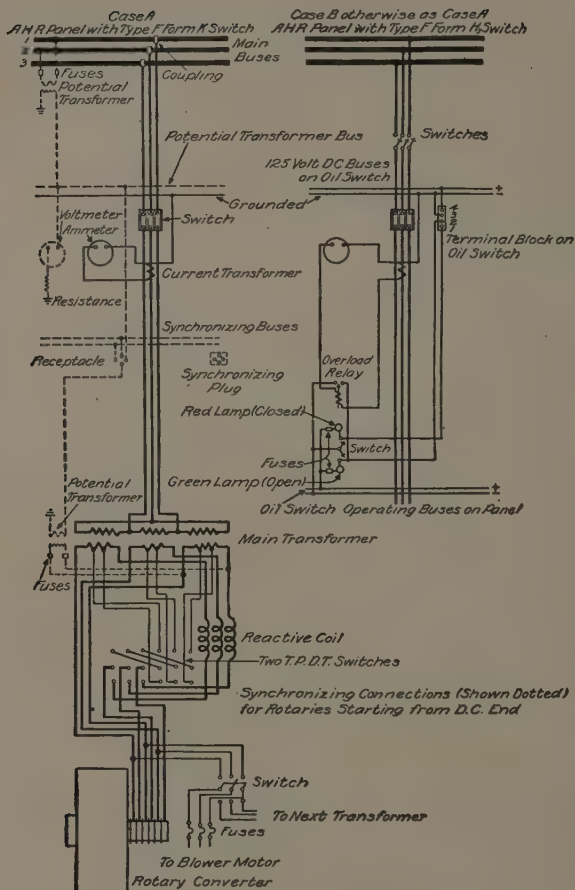


Fig. 329. A. C. End of Six-Phase Rotary.

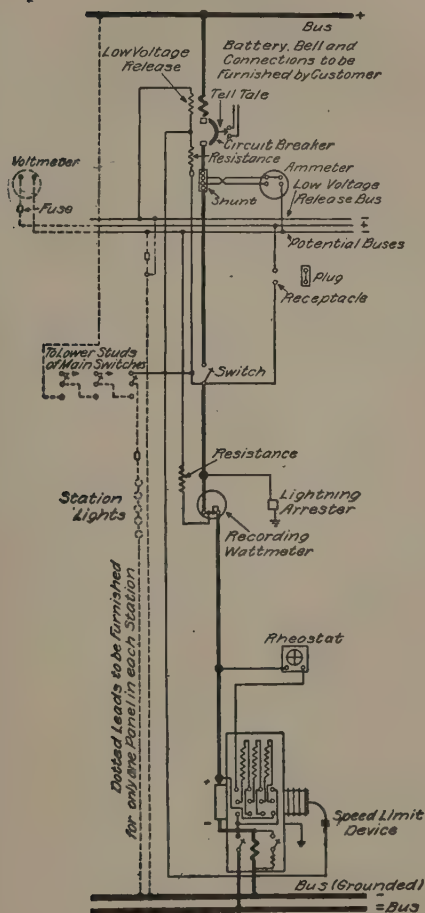


Fig. 330. Rotary Output Panel.

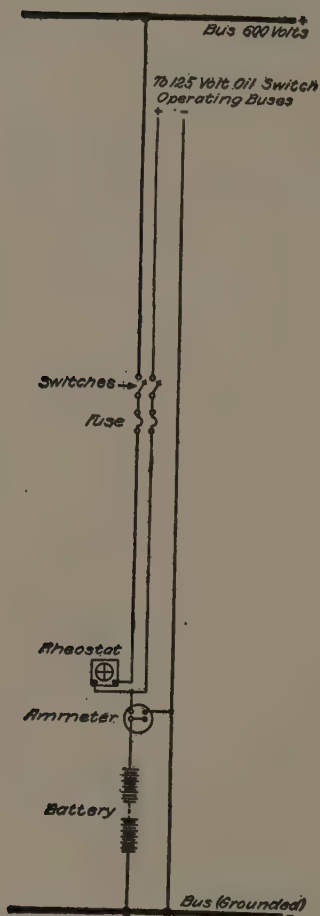


Fig. 331. Storage Battery Panel.

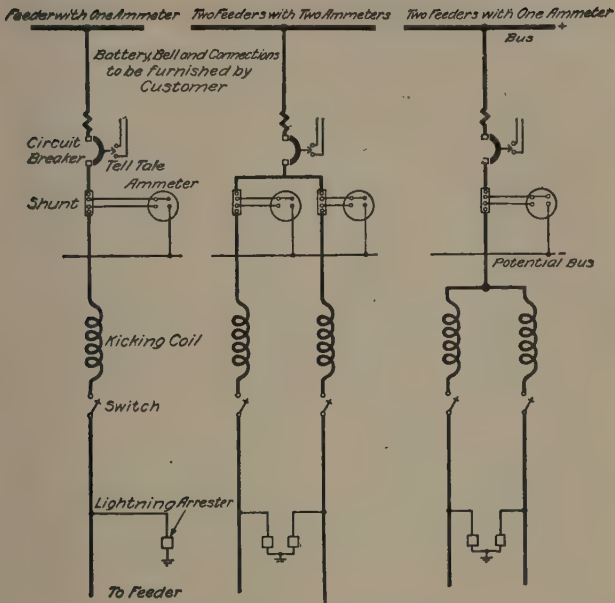


Fig. 332. D. C. Feeders.

are shown in Fig. 328, while similar panels for six-phase rotaries are shown in Fig. 329.

The connections of a rotary panel are given in Fig. 330.

The storage battery panel is shown in Fig. 331.

D. C. feeder panels are connected as shown in Fig. 332.

SWITCHES.

Lever switches of type shown in Fig. 333, are used only to close circuits. When used to open circuits the *Quick Break* type is used, shown in Fig. 288. Half the blade comes out of the clips when the handle is drawn back. This stretches the spring and the tension finally pulls the other half of blade out of clips at great speed, thus preventing much arc.

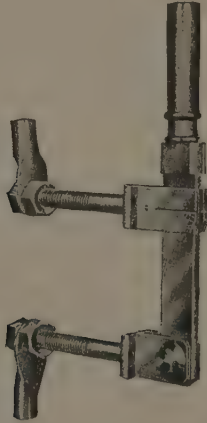


Fig. 333. Knife-Blade Lever Switch.

Field discharge switches, Fig. 289, are furnished only on the alternating current generator panels, and as they must be relied upon to shut down the generators under emergency conditions, they embody several features of importance. As stated in the description of the generator

panel equipment, the field switches are single pole and are furnished with clips for connecting a discharge resistance across the field when the latter is disconnected from its source of excitation, thereby preventing the generation of an excessively high potential, which might puncture the field insulation, when the field circuit is

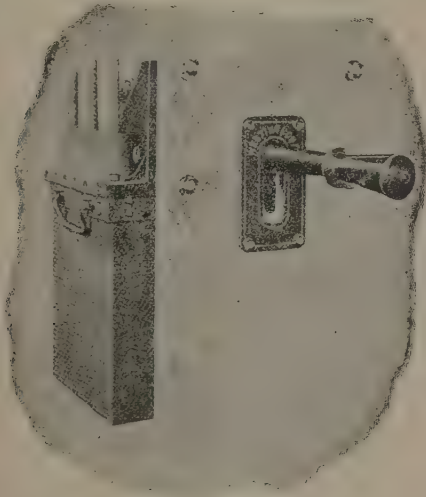


Fig. 334. Front View of Oil Switch.

opened. The construction is such that in opening the switch the resistance circuit is closed before the field is disconnected from the exciter, while in closing the switch the resistance circuit is opened before the field is connected to the exciter. By this means all destructive arcing is avoided, for the field can never be broken without shunting it through the discharge resistance, yet the lat-

ter is not even momentarily connected across the exciter bus-bars while the switch is being closed.

Oil switches. The severe conditions which must be anticipated on railway systems demand the use of switch-

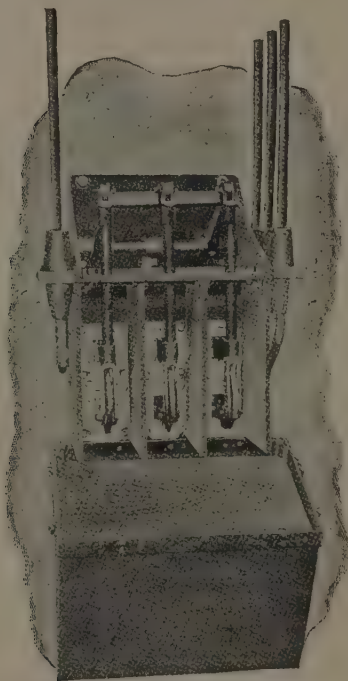


Fig. 335. Rear View of Oil Switch.

ing apparatus which will protect the machines and lines from excessive overloads and short circuits without injury to the switches, as the latter must be in condition to

put the machines back into service as soon as any temporary trouble has been remedied. These requirements are best met by reliable oil switches, properly installed in brick cells situated apart from the switchboard to prevent any local trouble from spreading.

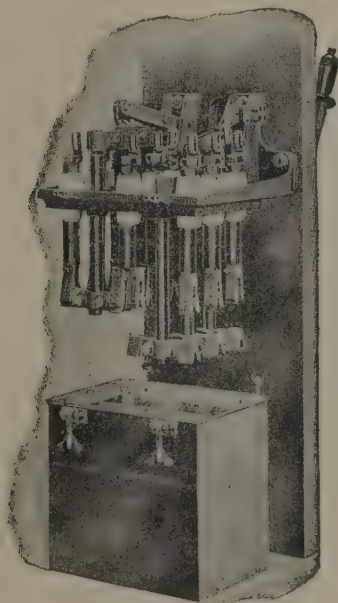


Fig. 336. Three Pole, Double Throw, Oil Switch.

In case of a dead short circuit on a feeder near the generating station, the current in the circuit which must be opened reaches a maximum, the value of which may be many times the normal capacity of the generators.

The automatic feeder switches must be capable of breaking this maximum current instantaneously. The non-automatic switches, on the other hand, are not required to open the maximum current, but must be capable of opening the load after the current has dropped to a value which the generator will maintain on short circuit.



Fig. 337. Oil Switch 15,000 Volt, 300 Ampere Triple Pole, Single Throw. Brick Partitions Not Shown.

Figs. 334 and 335 show the operating handle and the switch mechanism of a three pole oil switch.

A triple pole double throw oil switch for panel use is shown in Fig. 336.

It is often desired to have the switch at some distance from the panel on which it is controlled.

In Fig. 337 is shown a triple pole single throw 15,000

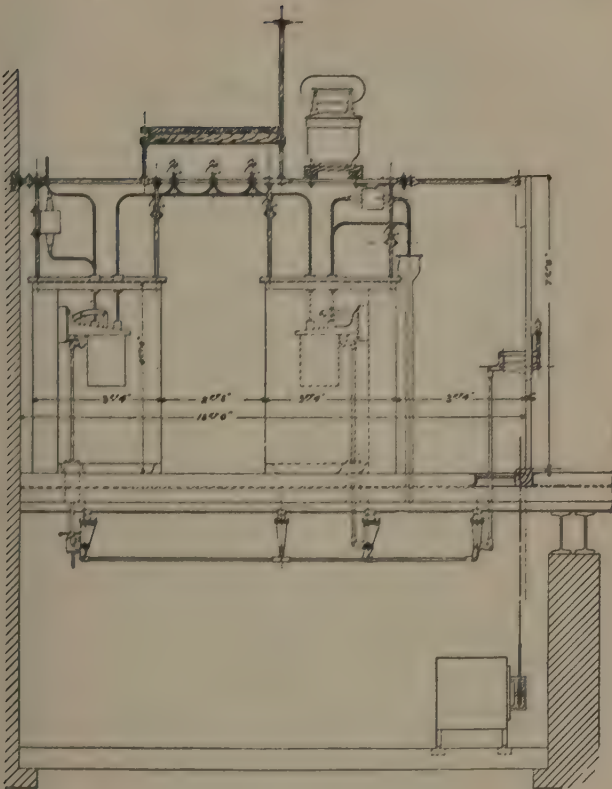


Fig. 238. Arrangement of Remote Control Lever Oil Switches.

volt 300 ampere oil switch. It would be very unsafe to have this on the switchboard.

It is built into a three compartment brick cell at some distance behind board and controlled by a long rod

running from lever on panel. This switch is of the non-automatic type, being used for a main generator switch.

Such switches, when used on transformers, are so far from the panel that some arrangement as in Fig. 338 is used.

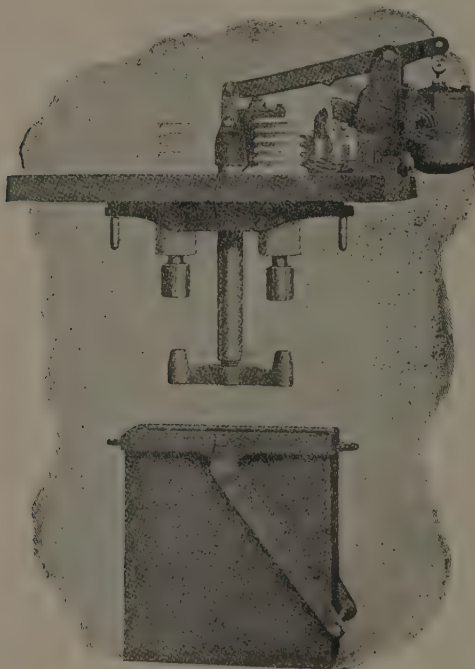


Fig. 339. Magnetically Operated Oil Switch.

Switches at a distance from the board may be controlled electrically if fitted with a tripping magnet and closed by sending an attendant to the switch. This is very bad practice, as the time wasted and even possibil-

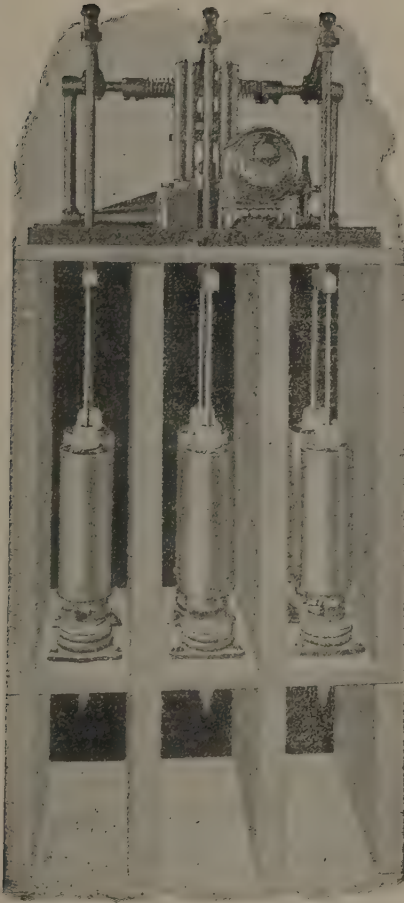


Fig. 340. Motor Driven, T. P. S. T. 13,000 Volts 300 Ampere Oil Switch in Brick Cell.

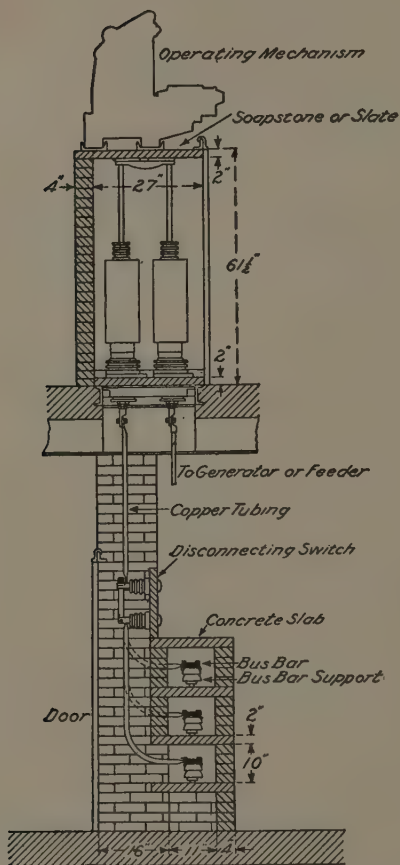


Fig. 341. An Arrangement of 13,000 Volt Buses and Oil Switches.
Very Little Floor Space Required.

ity of closing wrong switch must be considered. The switch in Fig. 339 is operated by two magnets, the left hand one (not clearly shown) opening, the right hand one closing switch. A small S. P. D. T. switch on panel

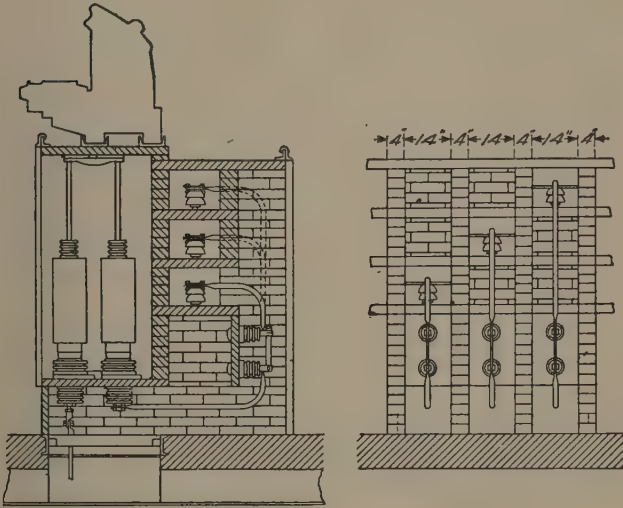


Fig. 342. A More Compact Arrangement Than Fig. 341. Needs More Floor Space.

controls the two magnets. Large switches, as Fig. 340, are opened and closed by motors. For such a motor switch there will be on the panel:

1 S. P. D. T. operating switch for the control of the oil switch motor.

2 miniature lamps, one with red and other with green bull's eye.

If the switches must open automatically there is provided in addition to the above:

1 overload relay.

The lamps with bull's eyes are controlled by an auxiliary switch attached to the operating mechanism of the oil switch. The opening of the oil switch lights up the green bull's eye, and its closing lights up the red bull's eye, assuring the attendant that the throwing of the controlling switch has been followed by the proper operation of the oil switch.

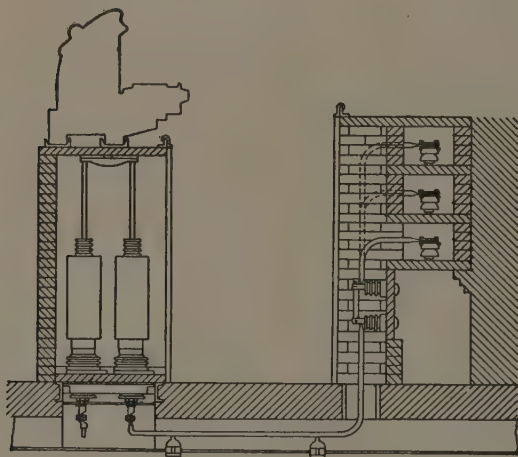


Fig. 343. Arrangement Similar to Fig. 342 When More Space Is Available.

When the current in the main circuit exceeds a predetermined value the overload relay closes a continuous current circuit, thereby tripping the oil switch.

Under some conditions, however, it is desirable to de-

lay, for a certain length of time, the opening of the switch under its predetermined overload. To accomplish this result the overload relay is equipped with a time limit device.

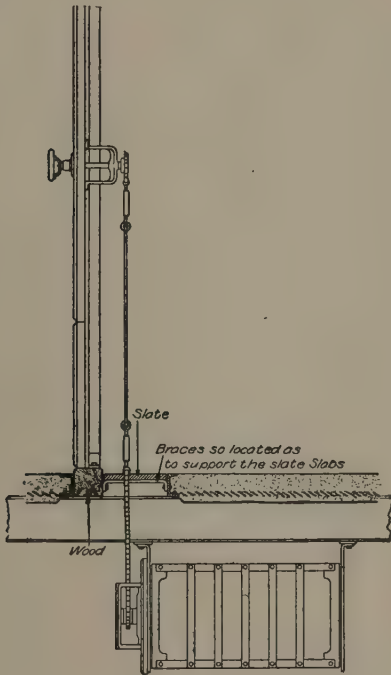


Fig. 344. Mounting and Control of Field Rheostat.

As already stated, all high tension oil switches should be mounted in brick cells situated apart from the panels.

The bus-bars, disconnecting switches and oil switches, need not only good insulation, but separation. Fig. 341

shows one way of handling a 13,000 volt set. Fig. 342 shows side and rear view of another scheme and Fig. 343 still another way of placing them.

Field rheostats being of such large size the arrangements shown in Fig. 344 are often necessary. In Fig. 338 the rheostat is on floor below.



Fig. 345. Clamping a Five Terminal Cable to Panel Stud.



Fig. 346. Voltage Detector, for Circuits Up to 15,000 Volts.

Cable connections. In order to connect large cables to the bus-bars, instrument, and machine terminals, the cables are split and the parts soldered into lugs. Fig. 345 shows how five of these lugs can be placed on a stud so as to occupy as little end room as possible.

The voltage detector, Fig. 346, is a static device having a movable element which is set in rotation when it is subjected to the line potential. A detector is attached to each conductor of every line, on the line side of the dis-

connecting switch. A complete set, therefore, consists of three detectors for each line. This device also serves as a ground detector, as its moving element will come to rest when a ground occurs on the conductor to which it is connected.

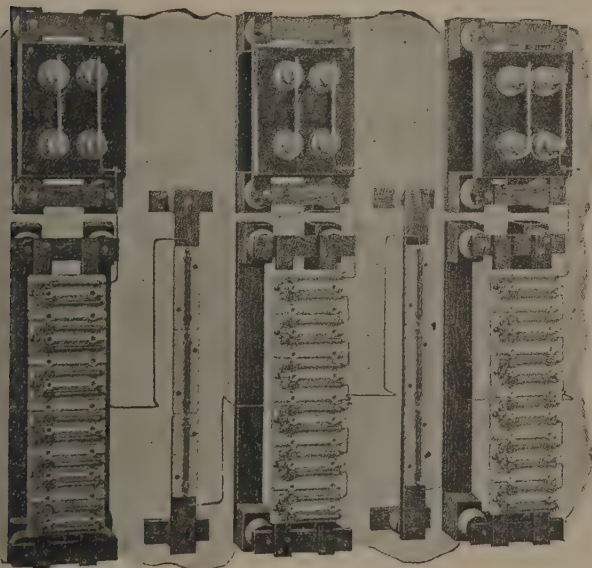


Fig. 347. 13,000 Volt, Three-Phase Lightning Arrester.

Three single-pole Double-blade disconnecting switches (Fig. 347) are placed in each incoming and outgoing line, one switch in each conductor. One blade is used for disconnecting the lightning arrester from the line, while the other blade is used for disconnecting the station from the line. Choke coils also are recommended for connec-

tion in series with the line between the arrester taps and the switchboard in order to protect the station apparatus. The arresters, switches and choke coils are designed for mounting on the wall at the point where the lines enter

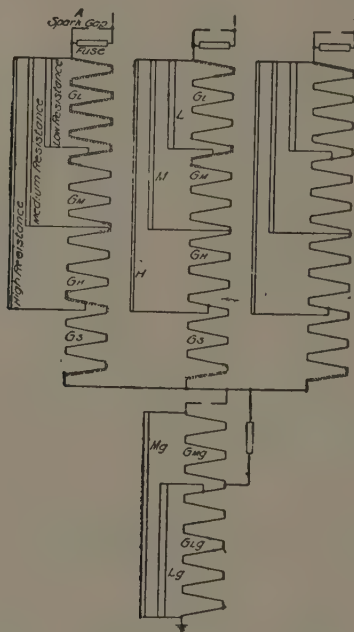


Fig. 348. Diagram of Multigap Graded Shunt Resistance, Three-Phase 33,000 Volt Arrester.

the station. The switches may be furnished either front or back connected, and are mounted on slate bases and provided with heavy corrugated glass insulators for the

switch studs. The blades are not provided with handles, but have holes in the ends to enable them to be operated by a hook mounted on a long wooden handle.

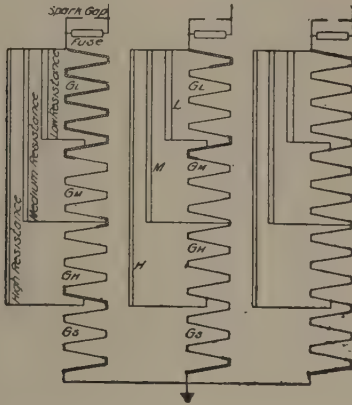


Fig. 349. Arrester Similar to Fig. 348 Arranged for Y System with Grounded Neutral.

Arresters. In Fig. 347 is shown a 13,000 volt 3-phase arrester with disconnecting switches. This is the older type, but is the kind in most stations.

In Figs. 348 and 349 are given diagrams of the connections of the newer graded shunt resistance type of arrester.

As considerable space has been devoted to oil switches, the subject will be continued still further by a brief explanation of the construction and operation of this type of switches, and also of oil circuit breakers. The principle of construction is shown in Fig. 350. On the right and left hand are two metallic rods which descend through insulating blocks, and carry springs at their lower end projecting therefrom. Another metallic rod descends between these two, and carries a cross-piece at its lower end, having beveled carbon contacts C. C. facing upward. This rod moves up and down through an insulating block, and it is connected to one lead of the circuit, while both side rods are connected to the other. When the central rod is raised the carbon blocks C. C. enter between the springs and make the contact, closing the circuit. When lowered it opens the circuit. Thus far the action of the switch is similar to the ordinary switch, but in order to prevent the formation of arcs, and to insure definite action in the opening or closing of the circuit, the lower portion of the mechanism is immersed in oil contained in a tank which is shown in section in the diagram Fig. 350. Oil switches are made in many different styles, but the one feature of the complete submersion of all live parts in oil, governs all. They are also as a rule held in the open position, either by gravity or by special locking mechanism consisting of a safety catch that holds the switch open until released by pressure of a button in the end of the operating handle.

The development of the oil switch and circuit-breaker has produced what is probably the most valuable addition to high-potential line apparatus made during the last ten years. It is indeed likely that the development of high-tension transmission of power would have been very seriously hampered but for the invention of the oil switch.

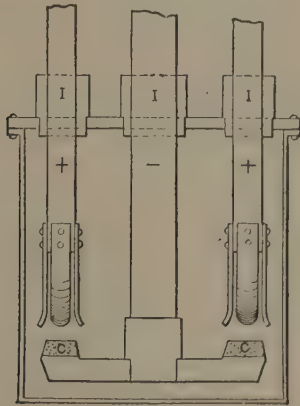


Fig. 350.

This use of oil has made it possible to rupture easily and safely circuits carrying heavy currents at high voltage and, further, to open these circuits under conditions of short circuit. The possibility of opening high-tension circuits under conditions of heavy overload has made possible the development and application of the present system of relays. By means of these relays, used in connection with oil circuit-breakers, perfect protection can be secured for the apparatus to which they are applied.

The term "switch" is given to those pieces of apparatus in which the contacts are similar to the ordinary switch, and are opened and closed by hand. Apparatus in which the contacts tend to separate, and are only held in a closed position by means of triggers and toggles are called "circuit-breakers."



Fig. 351. Type I Westinghouse Oil Switch.

The Westinghouse oil switches are essentially knife switches immersed in oil, the blades being connected to a common operating lever by specially treated wooden rods. The knife-blade contacts are used, as they give the most perfect contact and therefore the lowest temperature rise.

Fig. 351 shows a Westinghouse type I oil-switch for switchboard use only. Fig. 352 shows the type D Westinghouse oil-switch for outdoor service, the casing being moisture proof, and the leads brought out underneath through sealed bushings as shown. Fig. 353 shows the same switch with the oil tank removed. Fig. 354 shows

the Westinghouse type B oil circuit-breaker, designed for potentials from 3,300 to 22,000 volts, and currents from 300 to 1,200 amperes. This device is a double-break, oil



Fig. 352. Type D Westinghouse Oil-Switch.



Fig. 353

circuit-breaker. It may be automatic or non-automatic and may be placed on the back of the switchboard or arranged for distant control through rods and levers.

Through a very simple system of levers, the operating handle is connected to a cast-iron cross bar, to which are fastened the movable contact arms. To the lower end of



Fig. 354. Type B Westinghouse Oil Circuit-Breaker.

the wooden arm is fastened a metal yoke with a conical contact on either end. When the circuit is closed these contacts engage with two stationary contacts, forming one pole of the breaker. Each stationary contact is sup-

ported by a porcelain insulator, rigidly secured to the frame. The leads are brought to the terminals of the stationary contact within the insulator, forming an unbroken and continuous insulated conductor between the circuit-breaker and the bus-bar or line. Each pole of the circuit-breaker has a tank in which its live metal parts are immersed in oil, each tank being entirely independent of the adjacent one.

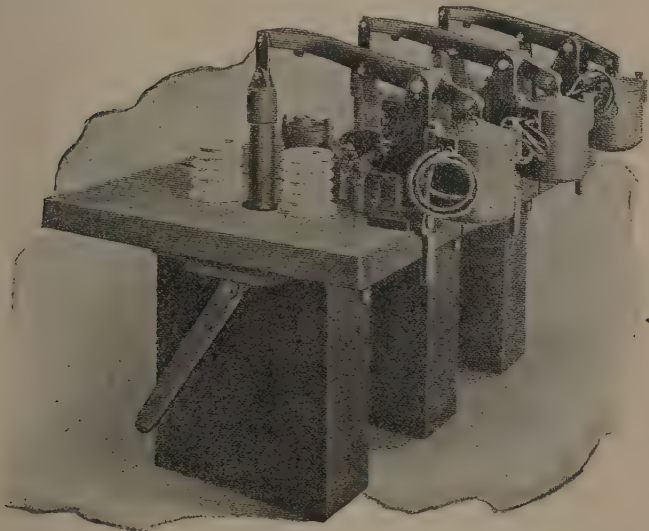


Fig. 355. Type E Westinghouse Oil Circuit-Breaker.

These tanks have a lining so formed as to reduce the quantity of oil required and which serves as a barrier between the two stationary contacts, yet allowing ample space for the movement of the wooden arm and its con-

tacts. The lining serves as an insulation and reduces to a minimum all danger of the arcs communicating through the oil. Any one of the tanks may be removed without interfering in any particular with the others. Fig. 355 shows type E Westinghouse oil circuit-breaker, electrically operated. This circuit-breaker is also designed for high potentials, and heavy currents.

A simple system of toggles and levers is mounted on the top of the breaker, and a powerful electro-magnet is arranged with its movable core attached to the lever system, so that when it is drawn into the coil, the circuit-breaker will be closed. A small single-pole, double-throw switch is mounted on the breaker, and is operated by the motion of the levers in opening or closing the circuit; it controls the tell-tale indicator, and lamp which are mounted in view of the operator. These circuit-breakers are operated by 125 volts direct-current, and are calibrated for 3,000 alternations.

SWITCHBOARD MAINTENANCE.

The increasing relative cost of switchboard apparatus in power plants justifies more thorough inspection on the part of attendants than at present obtains in many installations. There is a feeling in some quarters that if a switchboard is blown out every day with compressed air and the instruments wiped with a dust cloth, nothing further in the way of inspection need be done until something goes wrong.

There are more moving parts on a modern switchboard than one would at first suppose, and a certain amount of attention is an essential of continuous reliable service. In addition to the indicating and recording instruments

there are time limit relays, circuit-breaker controls, oil switch mechanisms and other contacts to look after, while the possibility of overheated parts of switches and coils is always present. Oil switches in operation should be inspected for overheating at least three times a day during the heaviest part of the load, and the binding posts of potential transformers, regulators and instruments should be looked after every two or three weeks with an eye to their becoming loose.

The oil tanks on oil switches ought to be dropped certainly once in three months and the contacts carefully examined to locate any broken or bent springs, burned contacts or loose connections. When these contacts are cleaned with a file or in any way where there is a chance of personal connection with the wiring system, the utmost care is essential that current should be cut off and high-potential contacts avoided. Knife switches for simple disconnecting work are worth many times their cost.

The solenoid equipment of time-limit relays are often neglected for long periods. The adjustment of these devices should be tested every two or three months and the contacts cleaned with the finest sandpaper or emery cloth. There is a tendency sometimes to forget that these relays are delicate apparatus. The adjustment of spring tension to hold contact pieces in place and the varnishing of solenoid plungers need to be carefully done. No little trouble can arise by careless varnishing of plungers so that they stick in one position and do not respond to the load variations above normal. Another point likely to be neglected is the care of the leather diaphragm on the relay bellows. This should be dressed with neatsfoot oil every two or three months to prevent it from becoming

stiff and hard. Lightning arresters should always be examined and placed in condition after a storm; rheostat contact points, fixed and movable, carbon brakes and copper feeder and switch jaws all need regular inspection just as much as commutators, brushes and bearings.

Instrument calibration is of less importance on a railway switchboard than in a central station selling current to individual consumers, but it would seem wise to check up the voltmeters and recording wattmeters at least once a year, and if possible to make an annual calibration check of the other instruments. The cost of power cannot be properly determined with recording wattmeters of inferior accuracy.

All these little details of operation are important factors in securing reliable service, and in some plants it will pay to reduce them to the compass of printed and framed rules.

METERS.

Galvanometers. An electric current passing near a magnetic needle deflects the needle, and if the current is passed first over the needle and then back under it in an opposite direction, the needle will be still further deflected.

An instrument consisting of a coil of wire carrying the current to be tested, and a magnet; the two being so arranged that one can be deflected, is called a galvanometer.

There are two types, the Thompson and the D'Arsonval.

The Thompson type has the coil of wire stationary and the light magnetic needle suspended by a silk thread. These can be made more sensitive than the other type, but are not portable and external fields have a great influence on them, causing them to give false indications. This is prevented by thick soft iron cases, much too heavy to be carried around.

The silk suspension makes the needle sensitive to vibration.

It is a fine laboratory instrument and with modified construction has been used in the workshop and field, but for this work the D'Arsonval is much better.

The D'Arsonval type has a very small light coil of wire suspended by a fine bronze wire between the poles of a stationary magnet. Since the movable part is not a magnet except during the actual instant of the test, outside magnetic fields have no influence on its motion.

To shield it during test a thin soft iron or steel case is put on the instrument which does not affect its portability. These covers are usually copper, brass or nickel plated for appearance sake, but the actual material is iron for the purpose of shielding the instrument.

These D'Arsonval galvanometers are not so sensitive as the other type, which for ordinary work is a great advantage.

Both of these types have the needle swinging over a circular scale divided into degrees, or may have a small mirror attached so that the deflection may be read by the motion of a spot of light moving along a long ruler supported about a yard away from galvanometer.

As mentioned before, twice the current does not give twice the deflection but by sending known currents through a galvanometer and marking a scale with pen and ink we could make an ampere meter. This is called Calibration.

AMMETERS AND VOLTMETERS.

The ammeters and voltmeters of commercial work are all special adaptations of the D'Arsonval galvanometer or for the least accurate work such as on switchboards; they are of the magnetic vane type.

The Weston instruments are D'Arsonval galvanometers.

Fig. 356 shows an instrument with the cover removed. A large permanent magnet of U shape is supported by a gun-metal casting screwed to the ends of the limbs, and the whole of the working part is built up on this magnet independent of the case, so that the movement can be removed bodily from the case by simply taking out one screw which holds the gun-metal casting in place.

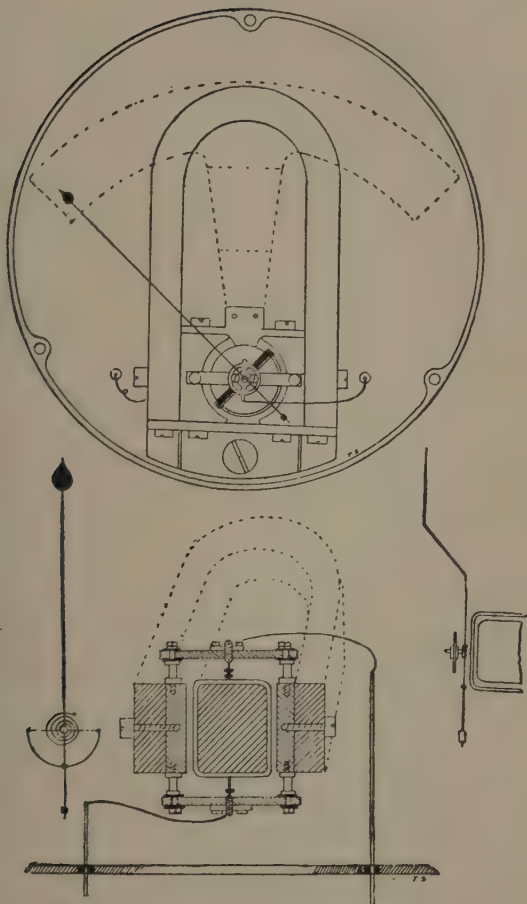


Fig. 356. Interior of Weston Instrument.

The inside polar faces of the magnet are surfaced up so as to come closely into contact with wrought-iron pole pieces which are bored out to about 1 in. diameter, and fixed rigidly in their place with screws passing through the limbs of the magnet. To these pole pieces a second gun-metal casting is screwed, which forms a support for a soft iron cylinder $\frac{3}{4}$ in. diameter inside the bored out pole pieces, and also a support for the scale. The soft iron cylinder fills up most of the space between the pole pieces, allowing an air space at either side of $\frac{1}{8}$ in., and in this space a fairly strong, uniform, magnetic field exists. A coil of fine insulated copper wire of about twelve turns is wound on a thin brass frame, large enough to embrace the soft iron cylinder, with freedom to move in the space between it and the pole pieces. This is pivoted in jeweled bearings which are screwed to the pole pieces, but insulated from them, forming little bridges across, and the ends of the coil are connected to these bridge pieces by spiral springs, one at the top and the other at the bottom of the coil, the springs being wound in opposite directions, so that when one is wound up by a movement of the coil the other is unwound. This arrangement corrects for any variation in temperature, for the effect on one spring would be counteracted by the opposite effect on the other. The coil normally lies at 45° to the line joining the poles of the magnet, and consequently the magnetic field created by a current in the coil will be displaced relatively to the field of the horseshoe magnet as shown in Fig. 357, and the lines twist the coil through a certain angle against the action of the spiral springs, the angular movement of the coil depending on the strength of the current in the coil and the strength of the field in which it is placed.

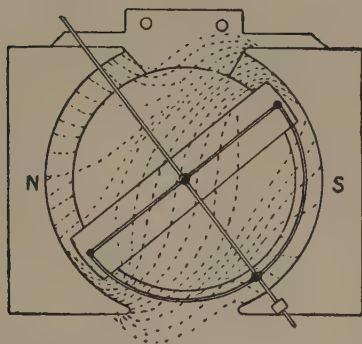


Fig. 357. Diagram of Magnets, Flux, Coil and Inner Core of Weston Instrument.

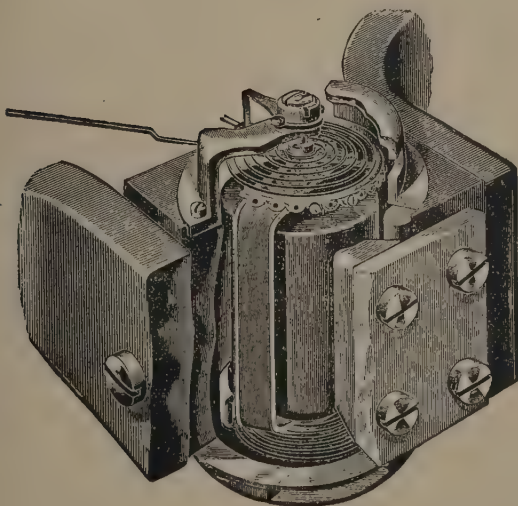


Fig. 358. View of Movement of Weston Instruments.

To the coil is attached a pointer of aluminum, the whole being balanced so that the instrument can be read in any position, and the pointer and scale are bent up so as to come near the front of the case.

A perspective view of the movement is shown in Fig. 358.

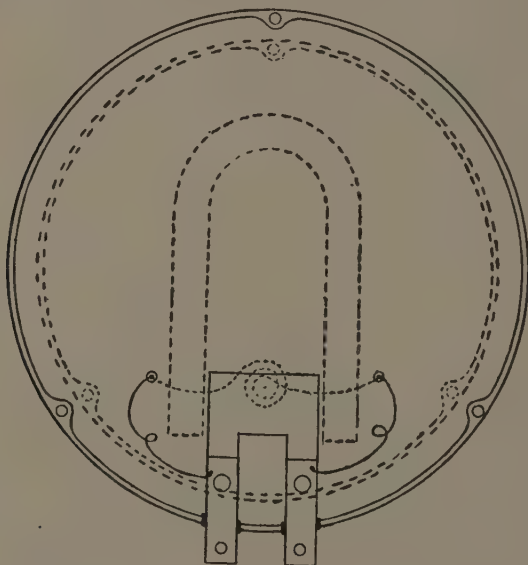


Fig. 359. Magnet and Shunt of Weston Ammeter.

In this instrument the whole current does not go through the coil, but only a small fraction of it. The main part of the current crosses from one terminal to the other by a broad strip of metal under the base of the instrument, while the coil is placed as a shunt across the terminals, or as a conductor in parallel with the

metal strip (Fig. 359), and consequently the ratio of the currents in the strip and in the coil will be inversely proportioned to their resistances. Now with a given strength magnetic field due to the magnet, and a given elasticity of the spiral springs, it will require a certain number of ampere-turns in the coil to produce the full deflection on the scale. This can be secured by adjusting the resistance of the strip connecting the terminals so that the same movement will do for any instrument. Thus, suppose the instrument were required to read to a maximum of 10 amperes and we required 1 ampere in the coil to give the maximum deflection, then the resistance of the coil must be 9 times that of the strip, so that the current will divide at the terminals, 9/10 going through the metal strip and 1/10 through the coil. If the instrument is required to read to a maximum of 100 amperes, then the metal strip must have 99 times less resistance than the coil, and the current will then divide at the terminals, 99/100 going through the strip and 1/100 going through the coil, which will give a reading to the full range of the scale as before. By the arrangement of the pole pieces, and wrought iron cylinder the field due to the permanent magnet is practically uniform over the range of movement of the coil, and so the scale readings are the same size throughout. Should the permanent magnet vary in strength, the instrument would not read correctly, but the magnets are so treated that the falling off in strength over a number of years is inappreciable.

The strip or shunt for portable instruments is always inside the case, while for switchboard instruments the shunt is too large (Fig. 360) and is placed separately on the back of the board. Leads are run along the board

to the meter terminals which project through holes in the board from the meter which is in front.

Such a switchboard instrument is shown in Fig. 361 and Fig. 362.

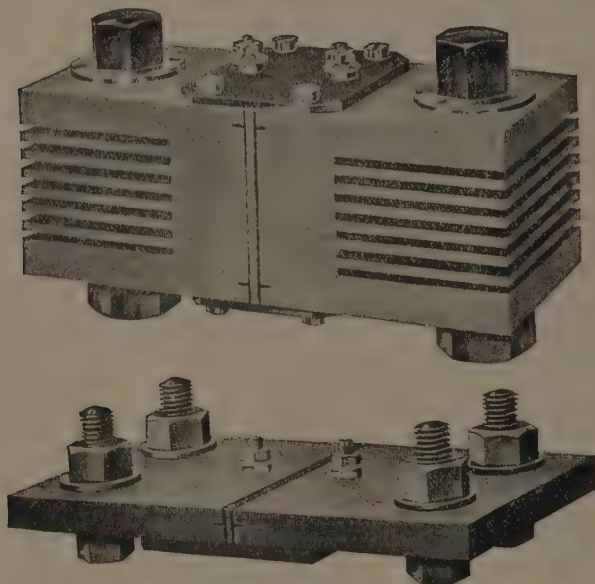


Fig. 360. External Shunts for Ammeters. 1000 and 5000 Ampere Sizes.

A voltmeter is made by removing the metal strip or shunt connecting the terminals and placing a resistance coil in series with the moving coil.

As it takes $1/100$ amperes to swing the pointer over full scale for every volt the instrument reads, it must have 100 ohms in the resistance coil.

A 500-volt instrument will have 500,000 ohms resistance, and hence $1/100$ amperes will flow through the moving coil.

The moving coil is wound on a copper or aluminum frame, which when it swings has current induced in it by the magnets and stops vibrating very quickly; in fact, you cannot detect any vibration. The needle seems to move to a certain spot and stop dead. This is called a "dead beat" needle.

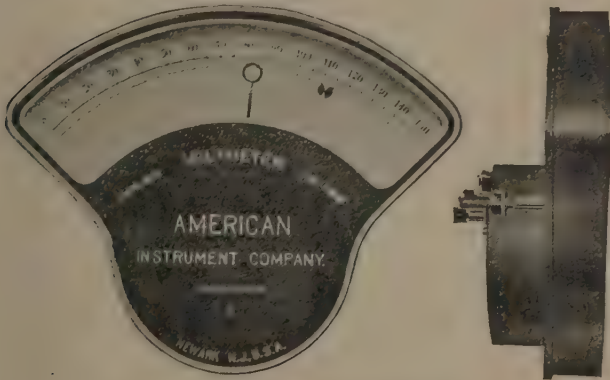


Fig. 361. Switch Board Instrument.

Some instruments have electro-magnets instead of permanent magnets. The Thomson Astatic instruments are of this type. Two of these instruments are shown in Figs. 362 and 363. This latter has a scale or dial of opal glass with an electric light behind it. This makes the instrument easily read from a distance or at night. These are called "illuminated dial instruments."

The instrument of Fig. 361 has an extra hand ending in a ring. This can be set at the voltage you wish to

maintain. The most hasty glance will then tell you whether your voltage is too high or too low.



Fig. 362. Switch Board Ammeter.



Fig. 363. Illuminated Dial Instrument.

In order to save space on switchboards some instruments are made thin and broad and are set horizontally or vertically.

Fig. 364 shows the exterior and interior of a Thomson Edgewise Ammeter.

The Thomson Inclined Coil instruments as shown in Fig. 365 are portable instruments used for alternating current only. In an emergency they can be used to measure direct current by reading, reversing current, reading again and averaging results.

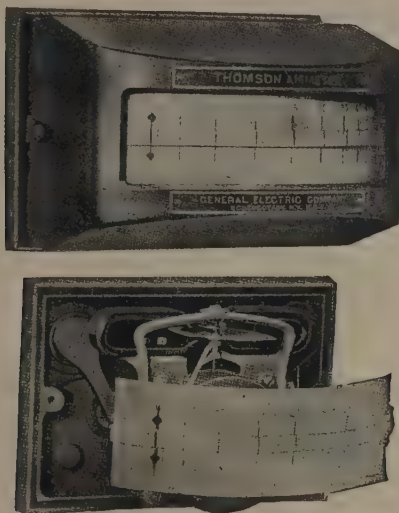


Fig. 364. Thomson Horizontal Ammeter.

The action of the magnetism of the inclined coil is to twist the inclined sheet iron vane "a" around to the dotted line position.

The Weston instruments described are for direct current only. The company makes an alternating current

voltmeter but no ammeter. Thomson Astatic instruments are for direct current. The Thomson Inclined Coil in portable, or edgewise switchboard form is for alternating current.

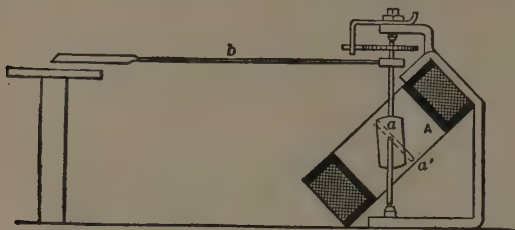


Fig. 365. Thomson Inclined Coil Instrument.

WATTMETERS.

By combining two coils, one movable, the other stationary, one attached as a voltmeter with series resistance, the other attached as an ammeter, with a shunt, we get an instrument whose needle indicates power or watts. These are called indicating wattmeters.

The recording wattmeter records watt-hours. A watt-hour is one watt of power used for an hour, or any combination like one-quarter of a watt for four hours, etc.

The Thomson Recording Watt-Hour Meter is used for direct or alternating current. It is shown with dust-proof case removed in Fig. 366. The connections made to it are shown in Fig. 367. By "light" in the figure must be understood any load at all.

The meter consists of an electric motor whose armature *A*, Fig. 367, is supplied with current from the mains through a high resistance *P* in the back of the instrument and a small field coil on right-hand side.

This armature is in shunt across the circuit, hence its current is proportional to the voltage.

The main current passes through the field F , hence the strength of the field is proportional to the current.

The speed of the motor is therefore proportional to both current and voltage, that is to the power, or watts.

The armature shaft goes on past the commutator to a cyclometer with dials like a gas meter. The revolutions are here recorded as watt-hours.

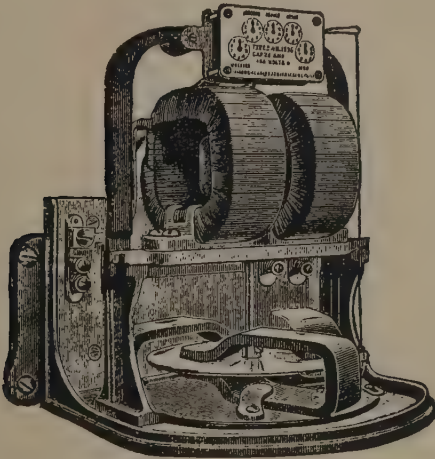


Fig. 366. Thomson Recording Watt-Hour Meter.

The auxiliary field is just strong enough to nearly overcome the friction in the bearings and cyclometer, so that the smallest current through the mains will produce rotation.

At the bottom of the armature shaft is an aluminum disk revolving between the poles of permanent magnets. This device prevents the meter from running at too great a speed and gives an adjustment for accuracy.

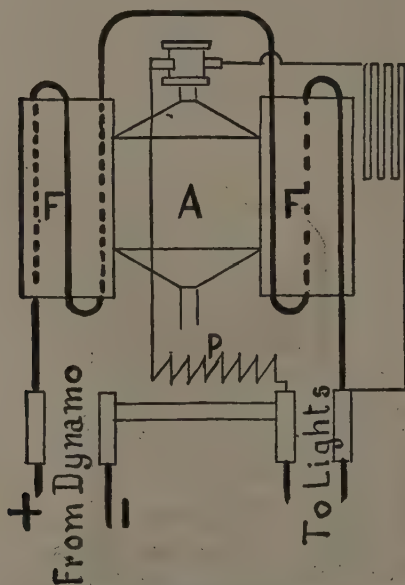


Fig. 367. Diagram of Connections. Thomson Recording Watt-hour Meter.

The further out the magnets are swung the faster is the motion of the metal passing between their poles and the greater a retarding effect they produce.

A meter running too slow from age or dirty bearings could be brought up to proper speed by swinging the magnets in a little.

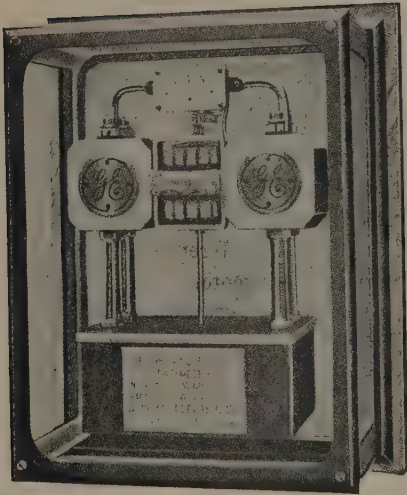


Fig. 368. Large Capacity Thomson Recording Watt-hour Meter.

For heavy currents the appearance of the meter is quite different, as is shown in Fig. 368. The retarding device is enclosed in a case and the whole instrument is enclosed in a dust-proof glass case.

QUESTIONS.

322. How are switch boards made up?

Ans. They are built up of panels of slate or marble supported by frames of angle iron.

323. How are the different panels designated?

Ans. Some are for motor control, others for dynamo running, others for operating the outer circuit, and others for charging storage batteries.

324. Is a knowledge of switch boards an important matter?

Ans. It is, and every engineer should especially study those in his own station.

325. What is the regular equipment of a D. C. switch board having a capacity of from 250 to 6,500 amperes?

Ans. One carbon-break or magnetic blow-out circuit breaker with telltale.

One illuminated dial ammeter with shunt.

One hand wheel and chain for operating rheostat.

One receptacle for voltmeter plug.

One S. P. S. T. field switch.

One S. P. S. T. main switch.

One recording watt-hour meter.

326. What is meant by the abbreviation S. P. S. T.?

Ans. Single Pole Single Throw.

327. What does D. P. D. T. mean in speaking of switch boards?

Ans. Double Pole Double Throw.

328. What is meant by T. P.?

Ans. Triple pole. It opens every circuit of a three-phase system.

329. Is it good practice to place a main switch at the machine?

Ans. It is best.

330. Why?

Ans. So that the cables from generator to board may be cut off at the generator.

331. What is an equalizer?

Ans. It is a cable running along from machine to machine, and connecting the functions of series field and brush on all the machines, but does not connect with switch board.

332. What kind of a break has the field switch?

Ans. A carbon break.

333. Describe the action of a field switch.

Ans. Just before it opens it makes contact with an extra clip and puts a resistance on as a shunt around the field coils.

334. If this were not done what would be the consequences?

Ans. The fields would act as a spark-coil and the insulation be damaged.

335. When it is desired to throw a generator in parallel with other generators already running what is the proper method of procedure?

Ans. First—Close main and equalizer switches near the machine.

Second—Close field switch on panel.

Third—Close circuit breaker.

Fourth—Insert potential plug in receptacle and regulate voltage.

Fifth—When proper voltage is obtained close the other main switch on panel.

336. What is meant by voltage?

Ans. Electric pressure, or potential.

337. What is a volt?

Ans. The unit of pressure.

338. What is a voltmeter?

Ans. An instrument that indicates the voltage.

339. What is an ohm?

Ans. The unit of resistance.

340. Give a brief definition of Ohm's law?

Ans. The electromotive force equals the resistance multiplied by current intensity.

341. What is an ampere?

Ans. It is the unit of volume, or quantity-time unit for measuring the rate of flow of an electric current.

342. What is a coulomb?

Ans. It is an ampere-second. A coulomb equals the flow of an ampere of current past a given point each second of time.

343. What is an ammeter?

Ans. An apparatus for measuring current rate.

344. What is the meaning of the word watt as used in electrical work?

Ans. A watt is the unit of work. It equals volts $\times$ amperes.

345. What is the function of the wattmeter?

Ans. To record the watt-hours of work.

346. What is a kilo watt (K. W.)?

Ans. 1,000 watts.

347. Expressed in horsepower what is one K. W. equal to?

Ans. 746 horsepower.

348. What is a field rheostat?

Ans. An apparatus for controlling the current output.

349. What is the function of a transformer?

Ans. To transform the current from a higher to a lower voltage, or from A. C. to D. C.

350. What is meant by synchronism of electric machines?

Ans. When the maximum value of the E. M. F. in each machine occurs at exactly the same instant of time, the machines are in synchronism.

351. What is meant by the exciter panel of a switch board?

Ans. It is the panel that is equipped with the necessary switches, etc., for connecting the small exciter dynamo with the other generators in the station.

352. What is a sub-station?

Ans. It is the connecting link between the transmission line, and the trolley wire or third rail.

353. When A. C. is generated at the power station, and D. C. is used on the line, how is it accomplished?

Ans. The A. C. is changed to D. C. by rotary converters at the sub-station.

354. What is meant by frequency?

Ans. The number of times the current reverses per second.

355. What is the usual frequency for railway motors?

Ans. 25 is the standard.

356. What is a frequency changer?

Ans. A machine which receives current at one frequency and delivers it at another frequency.

357. What apparatus is used in an A. C. to D. C. sub-station?

Ans. Step down transformers, rotary converters, and A. C. incoming and D. C. outgoing switch boards.

358. What is the proper procedure for placing rotary converters in service?

Ans. After the machine has been started from the A. C. ends, and builds up with the proper polarity, first close the equalizer switch (on machine)—second, close circuit breaker on panel—third, insert potential plug in receptacle and regulate voltage—fourth, when the proper voltage is obtained, close positive switch (on panel).

359. What will be the result if the rotary builds up with polarity reversed?

Ans. The voltmeter will swing back of zero.

360. How may the polarity be corrected?

Ans. By means of the four pole, double throw field break up reversing switch mounted on the converter.

361. Describe an oil switch.

Ans. It is a switch similar in its action to other switches, with the exception that its mechanism is immersed in a small tank of oil.

362. What advantage is gained thereby?

Ans. Reliability of action in opening or closing a circuit.

363. Mention another advantage gained by the use of the oil switch and oil circuit breaker.

Ans. It has made safely possible the transmission and use of high-tension currents of electricity.

364. What is a circuit breaker?

Ans. It is a switch so designed as to be capable of frequently opening the circuit carrying its full current without any damage to itself.

365. What is a galvanometer?

Ans. An instrument consisting of a coil of wire car-

rying the current to be tested, and a magnet, the two being arranged so that one can be deflected.

366. Describe the Thompson type of galvanometer.

Ans. The coil of wire is stationary, and the light magnetic needle is suspended by a silk thread.

367. Describe the D'Arsonval galvanometer.

Ans. In this type the small light coil of wire is suspended by a fine bronze wire between the poles of a stationary magnet.

368. How are the readings taken from these instruments?

Ans. From a circular scale, over which the needle of the instrument swings.

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